POWER

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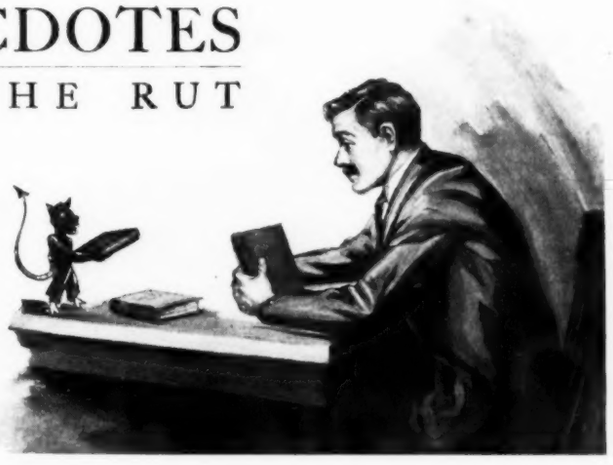
PURPOSEFUL ANECDOTES

WHY JOE STAYS IN THE RUT



All have heard of Dr. Eliot's five-foot shelf of books, which, it is claimed, offers the means for acquiring a liberal education. Happy indeed is the man that seeks this knowledge—happy in the compendiousness of his library, the conciseness of the text and the learnedness of his preceptor.

Were we to have a counselor equally as honest, equally as profound, who would guide us to a small shelf of books offering a true education in our trade, how blessed we should be, none save he who has experienced the trials of Joe Hently can fully appreciate.

LISTEN TO JOE'S STORY

"YOU ask why I don't study and climb out of this rut I'm in. Well, the book-bug does buzz about my ears some times, and once it stung dinged hard. It was when young Cullen strided over my head. I got real furious that time and vowed to turn over a leaf and get scent of some learning.

That same evening I sets out for the city library and asks the usher to show me some books for engineers.

"'I aint particular,' I tells him, 'so long's 'taint nothing that wants more than the three R's in education.

"'Choose for yourself, sir," says he, and he brings out a couple of drawers full of little tickets with lots of engineers books printed on; I never saw such a many before. 'Surely,' thinks I, 'some out of the lot must suit me.'

"Well, I spends an hour over them tickets, and finally I picks three that bore names that promised something easy for a starter. Soon my books were planked down before me and I plunged into the topmost, and sure enough, I had picked out an easy one—so tarnation easy as to be fit to be read only by a lubber that wouldn't know a boiler if it exploded under him. Weren't that the luck?

"Well, I had two others, but I picked up the next one with a lot of ardor knocked out of me, for I surely thought that mate Jinks, my special hoodoo, had trailed me. But my spirits rose with the reading of this one, for it was much more to my liking and I soon got mighty interested and thought that now, at last, I was going to learn something. I found a tidy explanation of a rule that had been the bane of me; and I determined to copy it and practice it in the engine room. The young fellow gave me some paper and a pencil, and I was deep in the strategems of the rule altogether unconscious of mate Jinks, when, bang! he bumped me into the middle of a bunch of sines and cosines, and other mathematical stuff that was all Greek to me.

"So, I put that book by with big regrets, and I was afraid to pick up the other—I couldn't stand another disappointment. But as I stared dubiously at it, Jinks picked it up and handed it to me. Of course, I wasn't a bit more vexed when it turned out to be nothing but catalog stuff. I got up from the table, thanked the young fellow, refused some more books and gave up the fight with old Jinks.

"Now you see why I don't study. I'm willing all right, but if I can't pick one book out of a thousand, what's the use? It's the same when I buy my books. I put up hard-earned cash for stuff that's fit only to light the fire with. It's a shame!"

Such is the sad story of Joe Hently. It is one of those things

> ".....of that nature as to make
> One's fancy chuckle, while his heart doth ache."

Does not his experience twang a responsive chord? Have we not met the same disappointments so often that it is our belief that buying an engineering book is like buying "a pig in a poke?" "Some books are to be tasted, others to be swallowed and some few to be chewed and digested," says Bacon. But how are we to distinguish that which is bread from that which is stone. Surely not by their titles, for that is almost as deceptive as judging them by their covers.

The United States Government guards the physical food of its subjects by a pure-food law. Would it could also guard the citizens' mental food.

One publishing firm allows five days' free examination of its books, thus tacitly acknowledging that titles, and even synopses are delusive, that engineers are often "stung," and that there are authors today, as there were in Cervantes' time, "that will make you books, and turn them loose into the world with as much despatch as they would do a dish of fritters."

A book is a book, they say. So is a horse a horse. But the trader that sells a defective animal is punished by law, unless he proclaims it unsound. Thus, the farmer is better protected than the student? Would that engineers' books might be labeled "good" or "bad," "complete" or "inadequate" by a competent board of critics?

Give us a Dr. Eliot to lead us to the good books; give us an honest man of mature judgment that will fearlessly point the finger of scorn at the worthless. Doubtless all the books required to liberally educate the engineer can be comfortably arranged on a two-foot shelf. But where are the books, who are their authors, who their publishers? IS THERE A DR. ELIOT to show us?

Contributed by R. O. RICHARDS, *Framingham, Mass.*

Regina Municipal Power Plant

By A. G. Christie*

SYNOPSIS—The particularly interesting features of this municipal plant are its boiler and economizer arrangement; its English turbines, condensers and other equipment, its motor-driven auxiliaries and its economical operation.

This article deals with some of the features of engineering interest connected with the new municipal electric-power station of Regina, Can., the capital of the province of Saskatchewan, which has a population in the neighborhood of 50,000, having grown in ten years from a town of 5,000 inhabitants.

The city first built what is now known as the old power plant in 1904 and added to it from time to time as was necessary. But the rapid growth of the city and the consequent tremendous increase in demand for electricity, as shown in Fig. 2, led the city commissioners to consider the problem of future plant extensions. These curves show that the average yearly increase for the last six years has been 50 per cent. of the previous year, while for the years immediately preceding 1914 it exceeded this figure considerably.

On account of the limited cooling-water supply at the old plant, a new site was chosen and machinery was installed during the fall of 1914 and spring of 1915, construction being still incomplete.

The plant is in the suburban section of the city on the shores of Lake Wascana. Plans and elevations are shown in Figs. 3 and 4. This section of country is underlaid with a deep bed of clay, which does not provide a firm base for engine foundations, but is spongy like rubber and allows considerable vibration to be set up if the engine parts are not in perfect running balance. However, if a heavy enough mass of foundation is provided, and especially if turbines are used, little trouble is experienced. The entire power house is consequently carried, as it were, on a raft of concrete resting on clay beneath, as a 2-ft. mat of concrete was laid over the bottom with reinforcing every 9 in. in both directions.

The main section of the building contains a boiler room 84x174 ft. and a turbine room 45x174 ft. The annex on the north side from the top down contains rooms for the air filters of the turbo-generators, for the switchboard, for switches and busbars and for feeder regulators and outgoing underground cables. A similar annex is on the south side.

Coal is purchased on specifications limiting the amount of ash and moisture. Before contracts are placed, sample carloads of the coals offered are delivered at the power plant and evaporation tests, each of 72-hr. duration, are run on the boilers and stokers. The coal company can have a representative present during these tests. During such trials all feed water is accurately measured in special tanks permanently provided for test purposes, and the coal and ash are weighed. Samples are taken of each for proximate analysis. Operating conditions are also carefully recorded. Good Western Canada bituminous coal costs about $5.50 per ton of 2,000 lb. at Regina. Souris lignite coal of low heat value can be obtained for $3 per ton.

Coal will be delivered to the plant in railroad cars and dumped into a steel hopper after weighing. The coal is automatically fed to a set of crushing rolls driven by a 30-hp. motor or in the case of slack, direct into the conveyor that elevates the coal into steel bins (capacity, 1,500 tons) suspended above the firing aisle, as shown in Fig. 4.

The coal feeds by gravity, through gates controlled from the boiler-room floor, into automatic scales that weigh out 500 lb. per dump. They are mounted over a hopper holding about half a ton, and all are suspended from a carriage which can be moved by a hand chain along tracks in front of the boilers. Two scales will be provided for each row of boilers.

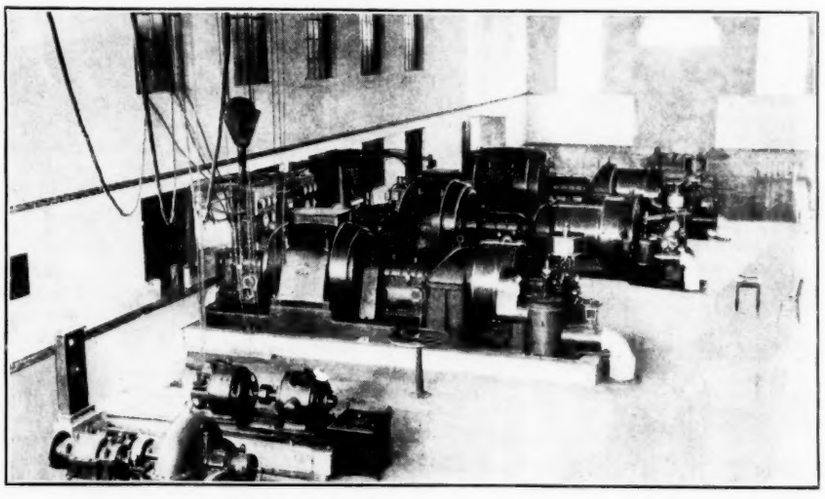


FIG. 1. GENERAL VIEW OF THE TURBINE ROOM

The same conveyor also handles the ashes, which fall from the stoker dump plates into brick-lined concrete pits with bottoms sloping toward the front. When a door is opened the ashes may be raked out onto a grizzly where the large clinkers are broken by hand before passing into the conveyor buckets. The ashes are elevated and weighed before dropping into a steel bin above the coal-unloading hopper and can be loaded from it into empty freight cars as desired.

On account of the induced-draft fans and flues, it was not possible to carry this conveyor to the extreme western end of the boiler room. Hence it has to be carried up through the boiler room at the end of the coal bunkers. This flight is inclosed, to reduce dirt, noise and the inflow of cold air, in a circular duct 4 ft. diameter and made of $\frac{3}{16}$-in. riveted steel. The spiral stairway leading to the upper conveyor ways and to the coal bunkers is carried around this duct, as shown in Fig. 5. This is quite convenient for the men.

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The boiler room, Fig. 6, now contains six water-tube boilers, each having 4,928 sq.ft. of heating surface, with superheaters containing 730 sq.ft. of heating surface per boiler. Two additional units are under construction. Room has been provided for a row of eight more similar boilers on the opposite side of the room. The operating pressure is 200 lb. per sq.in.

The economizer installation is one of the most interesting features of the plant. In place of the usual brick wall at the rear of boilers, a baffle extends from the floor to within 3 ft. of the level of the bottom of the drums. This baffle consists of two sheet-steel plates inclosing 2-in. asbestos blocks. An economizer for each boiler, consisting of 20 banks of eight tubes of 4 in. diameter and 12 ft. long, is placed just beyond this baffle and directly over the dampers to the flue below. The banks of tubes are set parallel to the boiler drums. The hot gases flow downward along the length of the economizer tubes into the underground smoke flue. The rear wall of the economizer chamber consists of a sheet-metal and asbestos curtain similar to the one between the economizer and the boiler. This is made in sections and easily removable for repairs. The economizer scrapers are driven by belts from a lineshaft supported by ball bearings and driven by a back-geared 10-hp. induction motor.

Recording thermometers are provided to register the temperatures of the gases before and after leaving the economizers. These temperatures range from 500 to 550 deg. F. leaving the boiler and from 300 to 350 deg. leav-

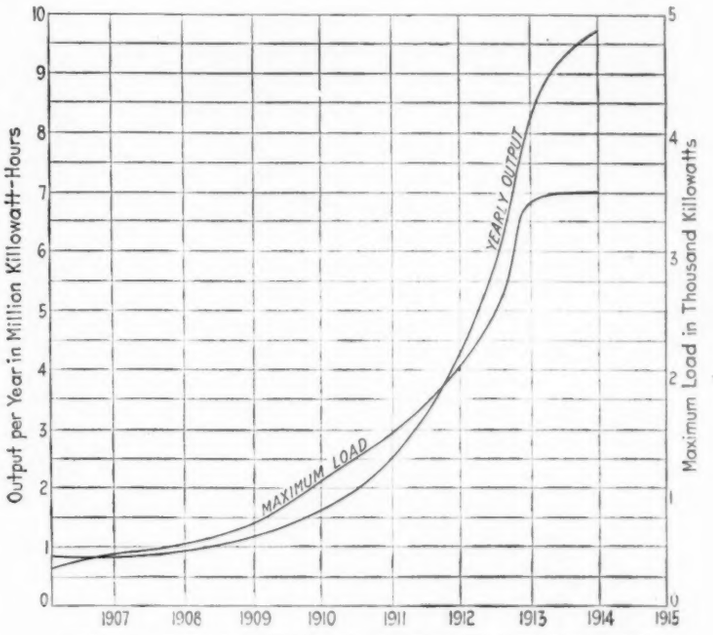


FIG. 2. CURVES SHOWING INCREASE IN MAXIMUM DEMAND AND YEARLY OUTPUT

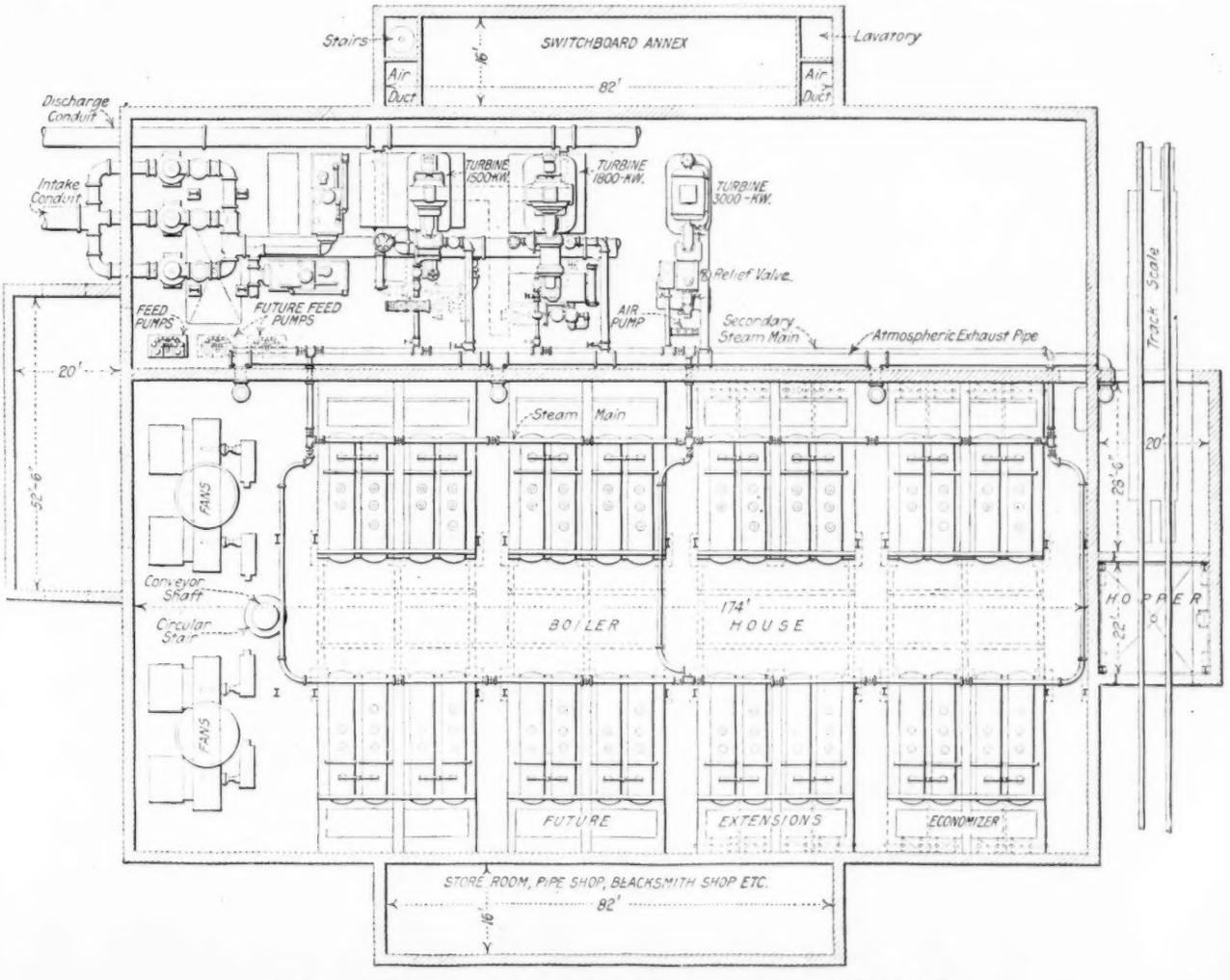


FIG. 3. PLAN VIEW OF THE REGINA MUNICIPAL POWER PLANT

ing the economizer, with normal load on the boiler and with 12 to 15 per cent. CO_2 in the flue gases. An unusual provision has been made to keep a spare boiler warm for emergency use. A damper has been placed in the central wall of the battery of boilers and at the bottom of the second pass. The smoke-flue dampers of both boilers are opened and sufficient flue gases pass through into the third pass of the reserve boiler to keep it always at working pressure. Both economizers are then used to feed the boiler in operation.

Each boiler is provided with a six-retort underfeed stoker. A lineshaft is run along the stokers slightly above the boiler-room floor level. The stokers are driven by gearing from this shaft. A multivane fan is suspended opposite the center of each stoker in the conveyor alley beneath the firing aisle. These fans are chain-driven from the shaft above. The fan draws its air from the conveyor alley. The hot air discharged from the turbo-

The gases leaving the economizer pass down into a concrete flue 8 ft. wide and 13 ft. high, with an arched top, located below the economizers. Draft is induced by either of the two multivane fans of 133,000 cu.ft. per min. capacity when working with a suction of 2 in. of water at the fan inlet. Each fan, Fig. 5, is driven by a 10x12-in. horizontal engine with a maximum speed of 240 r.p.m. and with automatic regulation. The fans are equipped with dampers on both sides and deliver into a stack 11 ft. diameter and 64 ft. high above the boiler-room floor. This stack is lined throughout with "vitrobestos" to prevent corrosion. Foundations for a similar induced-draft installation are in place on the opposite side of the boiler room.

An air heater has been built around the casing of the induced-draft equipment. By means of a variable-speed motor-driven fan and suitable ducts, air from the dressing rooms, washrooms, offices, etc., is forced through this

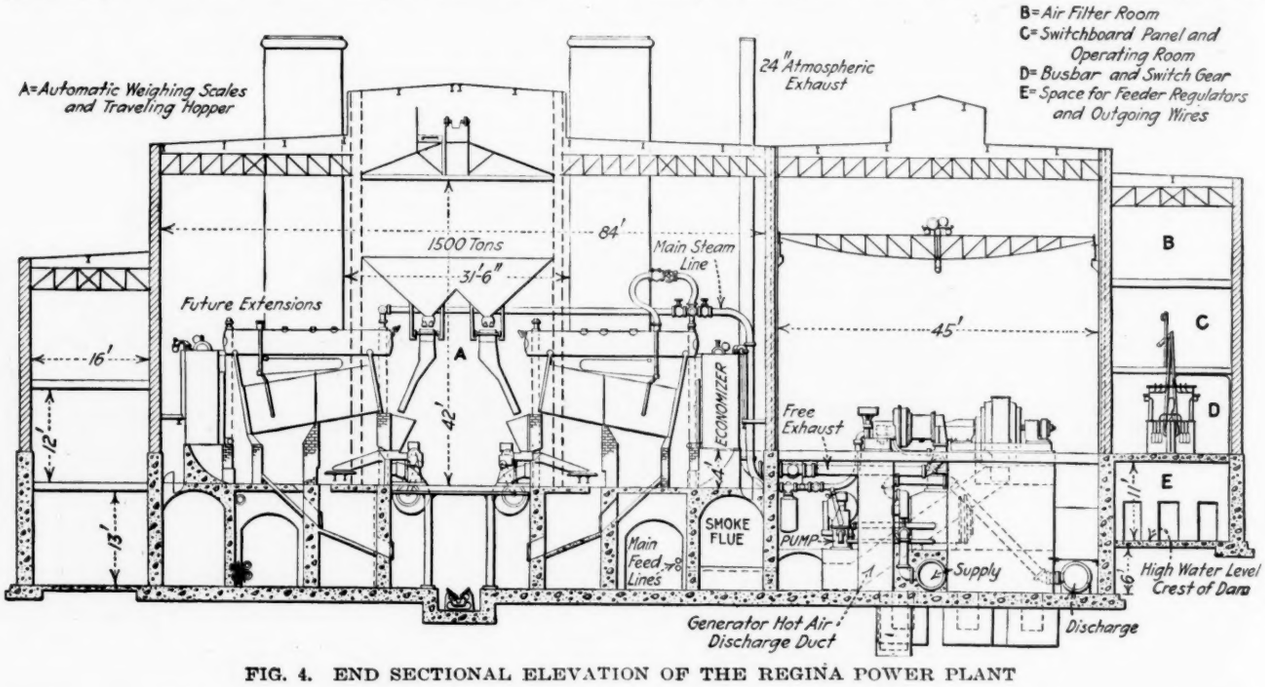


FIG. 4. END SECTIONAL ELEVATION OF THE REGINA POWER PLANT

generators is led by galvanized-iron ducts into this basement and together with the heat from the ashpits, etc., provides an air supply seldom below 100 deg. in temperature, even in winter. This utilization of waste heat tends to increase the over-all plant economy.

The stoker shaft may be driven by a 9x10-in. vertical engine controlled by a speed regulator, the maximum speed being 350 r.p.m., or by a variable-speed induction motor of 150 hp. and wound for 2200 volts.

A CO_2 recorder is placed in a prominent position in the engine room. A continuous record is made of the CO_2 from the boiler or boilers in operation. A system of lights, mirrors and ducts has been installed on the periscope idea, enabling the fireman to watch the operation of this recorder from the floor in front of one of the boilers.

The furnaces are provided with an 8-in. firebrick lining. The boilers have the usual vertical baffling. After a careful set of tests with Western Canada bituminous coal, it was found that the most economical results were obtained when a positive pressure of about 0.05 in. was carried in the furnace, and drafts are regulated to maintain this condition under all loads.

heater, where it is warmed by the flue gases, and is then delivered back into the rooms.

Feed water may be supplied to the boilers through either of two headers, of 5 in. diameter, ring-connected and equipped with gate valves for cutting out any section. Water may be fed to the boilers either by the regular connection into the economizers or by an emergency connection into the drum of the boiler itself. These feed headers are carried the full length of the boiler room in the basement alley along the side of the flue-gas duct. On account of the high temperature of the air in the boiler-room basement (about 100 deg.) and the moderate temperature of the feed water (also about 100 deg.), it was not considered worth while to cover these mains with insulating material.

The feed pumps are of the vertical type. The steam cylinder is 12x24 in. and the water cylinder 9x24 in. These pumps are fitted on the water end with sheet-steel valves, which have operated so far quite satisfactorily and show no appreciable wear. The pumps are in the basement of the turbine room. Their suction sides are connected to the discharge of a feed-water recorder built in

the heater, and thus a continuous daily record of water supplied to the boilers can be obtained.

A feed-water heater is in the basement of the turbine room, and all steam-driven auxiliaries exhaust into it. A water-storage capacity of 450 cu.ft. is provided in the heater, which is capable of handling 200,000 lb. of water per hr.

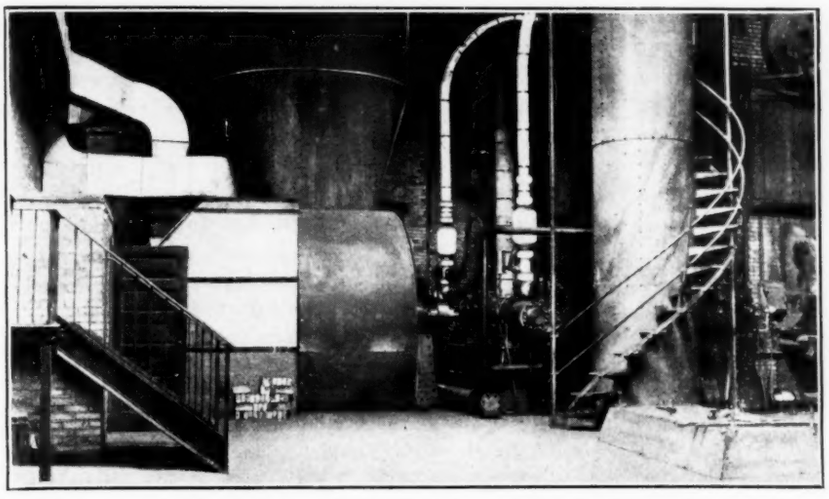


FIG. 5. INDUCED-DRAFT EQUIPMENT AND ENCLOSED CONVEYOR

Motor-driven auxiliaries are used wherever advisable; hence there is little exhaust steam available for heating the feed water, and the temperature leaving the heater seldom exceeds 100 deg. when all the main units are in operation. All drips, separator discharges, etc., are carried back and discharged into the heater, which serves principally as a receiver for the condensate from the turbines, at the same time utilizing all available exhaust heat. This method of operation provides practically a continuous closed system for all the water used in the plant, and the makeup water drawn from the city mains seldom exceeds 1,500 gal. per day. The discharge from the oil separator in the heater is collected, and the oil is allowed to settle out. This recovered oil is used on the bearings of the stokers.

Blowoff pipes from the boiler mud drum, the superheater and the economizer connect into a main blowoff header in the boiler-room basement, and this is carried underground to a concrete blowoff tank outside the power house. The scum is allowed to flow into the lake, the mud settles to the bottom, and the clear water is drawn off to either of two reserve water-supply tanks, which have a total capacity of 90,000 gal. The tanks are located in the basement. Rain water is sometimes collected in these tanks.

Steam superheated about 125 deg. leaves the boiler through 7-in. double pipe bends in form resembling a large question mark and passes into a 10-in. main header above the rear end of the boilers. A stop valve and an automatic nonreturn valve are located at the highest point of the bends, so that condensation drains away on either side and affords no opportunity for water hammer. The cross-section elevation, Fig. 4, shows this piping.

The main header is provided with sectionalizing gate valves. It is anchored in the middle and can expand in either direction on rollers placed between a supporting plate on the boiler frame and a saddle on the pipe. Two double-bend pipes of 10 in. diameter connect the header with a 10-in. secondary header in the turbine-room basement. Each pipe is provided with an ample separator just where it enters the secondary header. All live-steam pipes throughout the plant are covered with 3 in. of sponge-felt asbestos. The joints of all superheated-steam lines are made up with soft-iron gaskets. Since all connections to the headers are in the form of bends, there are no expansion joints; the only expansion joint in the plant is on the blowoff header, necessary on account of structural difficulties. A 6-in. saturated-steam header serving all steam auxiliaries is also placed above the boilers. This connection also has a gate valve and an automatic nonreturn valve between the header and the boiler.

The turbine room, as shown in Fig. 1, contains three main generating units and two exciter units. Provision has been made for three more main units of large size and for an additional motor-generator exciter set now being installed.

No. 1 turbine, delivering 1,500 kw. at 3,600 r.p.m., is one of the latest English turbine developments, having been installed in 1914. The rotor consists of a disk at the high-pressure end carrying a set

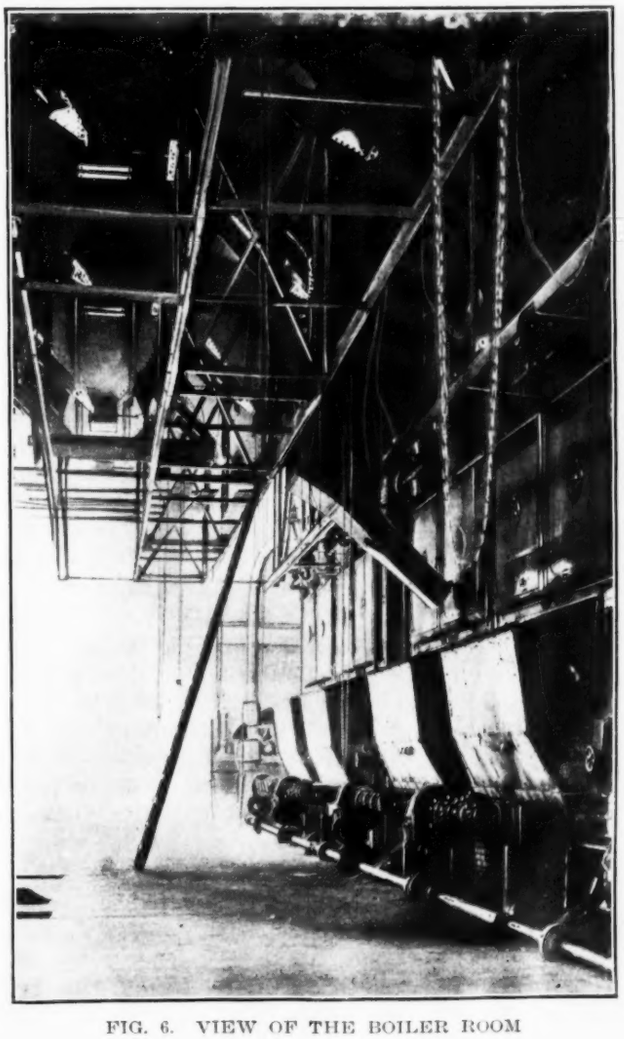


FIG. 6. VIEW OF THE BOILER ROOM

of Curtis blades supplied by steam nozzles in the casing. Then come the regular intermediate and low-pressure sections, of Parsons reaction blading. This spindle has a single balance piston on its high-pressure end.

No. 2 turbine was removed from the old plant where it was first installed in 1912. Its capacity is 1,500 kw. at 1,800 r.p.m.

No. 3 turbine was installed this year. Its normal output is 3,000 kw. at 3,600 r.p.m. Both No. 2 and No. 3 are provided with balance pistons. The design of No. 3 has been so improved over either of the other two units that it has been found more economical to operate it alone at any load over 800 kw. up to the capacity of the unit, rather than to use either of the smaller machines.

The average turbine-operating conditions are 180 lb. gage throttle pressure, 100 deg. F. superheat and 28 in. or more of vacuum referred to a 30-in. barometer. The vacuum is usually determined by a thermometer in the exhaust rather than by gages.

Oil coolers are placed in the basement. On account of the large volume of cooling water a pipe is led to each from the discharge of the circulating pumps and the water is carried back by another pipe to the suction of the circulating pump. As the rise in temperature and the volume of water required are small as compared with the condenser supply, they have no appreciable effect on the operation of the condenser.

The Turbo Generating Units

All turbo-generators are three-phase alternators wound for 2,200 volts. Nos. 1 and 2 turbines are provided with 17-kw. direct-connected exciters. Air for cooling is first drawn through cheese-cloth filters placed in a room above the switchboard where dust and insects are removed; it then passes down through the generators and is discharged into the boiler-room basement.

Each turbine exhausts into a separate surface condenser. No. 1 has 3,300 sq.ft. of cooling surface; the condenser is fitted with a vacuum augmenter. A three-throw vertical reciprocating air pump with a condensate pump on one end of the shaft maintains the vacuum and is driven by a 10-hp. induction motor. No. 2 turbine has a condenser with 4,400 sq.ft. of tube surface without augmenter and with a two-throw vertical wet-air pump, also driven by a 10-hp. induction motor. No. 3 turbine has a 6,650-sq.ft. surface condenser with a vacuum augmenter, and is provided with an air pump similar to that on No. 1, only larger and requiring a 20-hp. induction motor. The condensers with the vacuum augmenters give considerably higher vacuum and better economy than the one without it.

When operating noncondensing, the steam passes through an automatic relief valve into a 24-in. exhaust-steam header located under the turbine-room floor. A sheet-steel expansion joint is placed between the turbine and a special gate valve above the condenser which has a side outlet for noncondensing. The exhaust header is connected to the feed-water heater and is provided with two 24-in. outlet pipes in the boiler room, running to the roof. These pipes were used to avoid carrying a large atmospheric exhaust pipe through the engine room. Two additional pipes will be installed later.

The condenser cooling water flows through a 4-ft. concrete conduit from the lake to either of two 24-in. vertical centrifugal pumps. Each has a capacity of 12,500 gal. per min. against a 22-ft. head and is direct-driven by a 110-hp. induction motor running at 265 r.p.m. These pumps are below the lake level, and as either provides more water than is needed with the present summer load, a small belt-driven centrifugal pump brought from the old plant and mounted on the base provided for a third pump is now being used. The highest temperature of inlet water in summer has been 72 deg.; in winter it seldom exceeds 40 deg. The discharge water leaves the plant in a similar 4-ft. concrete conduit and is carried some distance down the lake before being discharged.

Nos. 1 and 2 generators have their own exciters. No. 3 is at present supplied by a 25-kw. motor-generator set running at 1,800 r.p.m. A 125-kw. turbine set is under construction. The high-speed noncondensing turbine drives the generator at 900 r.p.m. through reducing gears. A second motor-generator set of the same capacity is being installed.

Rates and Costs of Electrical Energy

Current is sold for both light and power on the two-rate basis. A monthly base or service charge of 50c. per kw. of demand is made in both cases. The rate for lighting is 7c. per kw.-hr. for the first 300 kw.-hr. Consumption above 300 kw.-hr. costs only 6c. per kw.-hr. The power rate is 5c. per kw.-hr. for the first 300-kw.-hr., 4c. for the second 300-kw.-hr., 3½c. for the third 300 kw.-hr. and 3c. per kw.-hr. for all in excess of 900 kw.-hr.

Two-rate meters are frequently installed, and special low rates are quoted for consumption during off-peak hours in order to increase the load factor of the plant. A 10 per cent. discount is given on all bills for prompt payment, and further discounts are allowed under certain conditions.

With these low rates the electric light and power department was able to show a surplus on operation during the year 1914 of $84,063 after allowing for depreciation. Of this amount $62,107 was carried over into the debenture and sinking-fund account, and the balance, $21,956, was transferred to the revenue account of the city.

Fig. 2 shows that the power developed in 1914 amounted to 9,725,455 kw.-hr. The financial statement of the city for 1914 shows the expenditure of the electric light and power department in schedule 12, and an analysis of this statement shows the figures given in Table 1. Of the power generated, 9,315,355 kw.-hr. was sent out of the station. The meters accounted for 8,223,955 kw.-hr., indicating a distribution loss of 11.72 per cent.

Table 1 shows a plant cost of 1.94c. per kw.-hr. and a total cost of 3.25c. per kw.-hr. These figures, with the high coal cost, refer to the old plant, as the new plant was not completed in 1914. The increased economy of the new plant will be to some extent offset by greater

TABLE I. TOTAL COST OF GENERATING CURRENT IN OLD PLANT

	Total Cost	Cost per Kw.-Hr. Developed in Cents
Fuel	$131,565.43	1.35
Plant wages	41,023.17	0.42
Oil, waste and packing	4,067.22	0.04
Water	824.48	0.01
Repairs to plant, etc.	11,991.57	0.12
Total plant cost	$189,471.87	1.94
Other costs to system:		
Office costs, etc.	55,259.26	0.57
Depreciation	19,370.65	0.20
Sinking fund and debentures	62,107.93	0.64
Total production cost	$326,209.71	3.35

fixed charges until the demand and output are considerably increased, but records of five months' operation indicate that the total production cost will be slightly reduced.

The new plant represents a capital expenditure of $700,000 and has an ultimate capacity of 20,000 kw. The plant records show for two months' operation with the new large turbine and electrical auxiliaries, a total coal consumption of 2.58 lb. of coal per kw.-hr. produced by the generator.

TABLE 2. PRINCIPAL EQUIPMENT OF REGINA MUNICIPAL POWER PLANT, REGINA, SASK., CAN.

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
8	Boilers	Horizontal water-tube	500-hp	Steam generators	200 lb. pressure, induced draft, stoker fired	Babcock & Wilcox Co.
8	Stokers	Underfeed	6-retort	Boiler furnaces	Forced draft from No. 7 Sturtevant fan	Sanford Riley Co.
1	Stack	Steel vitrobestos lined	64 ft. above floor, 11 ft. diam.	Discharges gases	Two induced-draft fans, 2-in. draft	B. F. Sturtevant Co.
1	Conveyor	Buckets overlapping		Handle coal and ash	Motor-operated	Eastern Steel Co.
1	Crusher	Robbins		Coal crusher	30-hp. motor	Eastern Steel Co.
8	Bunkers	Suspended steel bin	1600-ton	Holds coal	Filled by conveyor	Eastern Steel Co.
1	Bunker	Suspended steel bin	3,200-cu.ft.	Stores ashes	Filled by conveyor	Eastern Steel Co.
2	Weighers	Automatic dump	500-lb.	Weighs coal	Dumps weighed coal into movable hopper	Avery Scale Co.
1	Heater	Cochrane, open	200,000-lb. per hr.	Heats boiler-feed water	Utilizes all exhaust steam	Harrison Safety Boiler Works
2	Feed pumps	Vertical	9x12x24-in.	Boiler feed	200 lb. steam pressure	G. & J. Weir
8	Economizers	Steel tube	2320-sq.ft.	Heats boiler-feed water	Water enters at about 100 deg. F.	B. F. Sturtevant Co.
1	Turbine	Disk and drum	1500-kw.	Main unit	180 lb. steam, 100 deg. superheat, 28-in. vacuum, 1800 r.p.m.	Willans & Robinson
1	Turbine	Drum impulse and reaction	1500-kw.	Main unit	180 lb. steam, 100 deg. superheat, 28-in. vacuum, 3600 r.p.m.	Willans & Robinson
1	Turbine	Drum impulse and reaction	3000-kw.	Main unit	180 lb. steam, 100 deg. superheat, 28 in. vacuum, 3600 r.p.m.	Willans & Robinson
3	Turbo-generators	Three-phase, a.-c.	Same as turbines	Generates current	Same speeds as turbines respectively	Siemens Bros. Dynamo Works
1	Condenser	Surface with augmenter	3300-sq.ft.	Condenses steam from 1500-kw. turbine	28-in. vacuum with three-throw vacuum pump	Willans & Robinson
1	Condenser	Surface with augmenter	6650-sq.ft.	Condenses steam from 3000-kw. turbine	28-in vacuum with three-throw vacuum pump	Willans & Robinson
1	Condenser	Surface	4400-sq.ft.	Condenses steam from 1500-kw. turbine	28-in. vacuum with two-throw vacuum pump	Mirrlees, Watson & Co.
1	Exciter	Motor-driven	25-kw.	Excitation for 3000-kw. turbine	125 volts, 1800 r.p.m.	Bruce Peebles Co.
1	Exciter	Turbine-driven	125-kw.	Excitation for main units	Turbine driven with gear reduction, 125 volts, 900 r.p.m.	Terry Steam Turbine Co.
1	Pump	Wet-vacuum		Maintains vacuum in condenser	Two-throw, vertical, motor-driven	Mirrlees, Watson & Co.
1	Pump	Dry-vacuum		Maintains vacuum in condenser	Three-throw vertical, motor-driven	Willans & Robinson
1	Pump	Dry-vacuum		Maintains vacuum in condenser	Three-throw vertical, motor-driven	Willans & Robinson
2	Pumps	Vertical centrifugal	24-in.	Circulate cooling water for condensers	Motor-driven	Canadian General Electric Co.
1	Switchboard	Gray marble	25 panels	Carries switches and meters	All low voltage on main switchboard	Canadian Westinghouse Co.
1	Motor-converter	Le-Cour patent	1200-kw.	Supplies power to street railway	In old power plant	Bruce Peebles & Co.
1	Crane	Hand traveling	25-ton	In turbine room		Whiting Foundry Equipment Co.

This is equivalent to a coal cost of about 0.7c. per kw.-hr. generated.

The new plant was designed by Superintendent of Light and Power E. W. Bull, who has been in charge for eleven years. It is operated under his direction and credit is due to him and his staff, and to City Commissioners L. A. Thornton and A. W. Pool, for information contained in this article.

The Average Coefficient of Friction of iron or steel shafting in horizontal bearings is about 0.07 for ordinary oiling and about 0.043 for continuous oiling. Where d = the diameter of the journal in inches, the space passed over for one turn would be

$$\frac{3.1416 \times d}{12} = 0.26\,d \text{ ft.}$$

hence the power absorbed by each journal bearing where

W = Total weight of shafting and pulleys, plus the resultant stress of belts, in pounds;
d = Diameter of the journal in inches;
T = Number of turns per minute;
h = Horsepower absorbed, then for ordinary oiling the formula is

$$h = \frac{0.07 \times W \times 0.26\,d \times T}{33{,}000} = \frac{WdT}{1{,}813{,}187}$$

and for continuous oiling

$$h = \frac{0.043 \times W \times 0.26\,d \times T}{33{,}000} = \frac{WdT}{2{,}951{,}699}$$

Inflammability of Gasoline Vapor and Air

In Technical Paper No. 115, issued by the Bureau of Mines, Messrs. Burrell and Boyd present the results of some interesting experiments on the inflammability of mixtures of gasoline vapor and air.

Two methods of determining the content of the gasoline vapor in air were employed. In one the mixture was introduced into an exhausted glass vessel, cooled to the temperature of liquid air, the air removed, and finally the partial pressure of the gasoline vapor was measured by a manometer attached to the liquefaction bulb. The other method consisted in burning the gasoline vapor in oxygen and, from the contraction and carbon dioxide produced, calculating the percentage of gasoline vapor.

When a 100-c.c. Hempel explosion pipette was used and the mixtures were ignited from the top, there was obtained at the lower limit complete combustion, the mixture lying between 1.9 and 2 per cent. of gasoline vapor. The other limit under these conditions was found to be between 5.2 and 5.3 per cent. gasoline vapor. The gasoline used had a specific gravity of 74 deg. Baumé. Under the same conditions, except that the mixtures were ignited from the bottom, there was obtained a value lying between 1.5 and 1.66 per cent. of gasoline vapor at the low limit. With the same grade of gasoline, with a 2,800-c.c. vessel, with ignition from the bottom by means of an electric flash produced by pulling apart two wires through which a current of 7 amp. was flowing, there was obtained a value lying between 1.4 and 1.5 per cent. of gasoline vapor. The high limit under these conditions lay between 6 and 6.4 per cent.

When the initial temperature is increased before ignition of the mixtures, the low limit is gradually decreased until, with an initial temperature of 400 deg. C., the low limit lies between 1.02 and 1.22 per cent. of gasoline vapor.

The range of complete combustion for mixtures of gasoline vapor and air is very narrow, and is between about 1.5 and 2.5 per cent. The amount of carbon dioxide produced reaches a maximum at 2.5 per cent. of gasoline vapor.

At this point, as the percentage of gasoline vapor in the mixture increases, carbon monoxide begins to form. At 4.1 per cent. of gasoline vapor, there is produced 1 per cent. of carbon monoxide.

Visits of Inspector Brown--XII

By J. E. Terman

SYNOPSIS—Brown questions the Chief regarding spacing stay-bolts and discovers that there is more to the subject than most engineers realize. The Chief makes sketches of stay-bolt spacing and gives a few formulas.

When the Chief and Brown had gotten comfortably seated in the train, Brown said: "I want to ask you how to figure stay-bolts for boilers. I know in a general way how to get the direct stress on them, but I am lame when it comes to figuring the spacing. Although I have seen various rules for getting the maximum spacing between stay-bolts, I cannot make any sense out of them.

"A few days ago I inspected a small vertical boiler and had to fix the allowable pressure on the spacing between the bolts on the furnace sheet. The owner was an inquisitive fellow and wanted to know why I could not allow more pressure. I told him that the stay-bolts were spaced too far apart, but he was not satisfied and wanted to know how I knew. Well, I was right up against it and had to show the fellow my table giving the allowable pitch for stays and had to admit that I did not know any more about the subject than to follow the table. After seeing the printed table, the owner was satisfied that I could not allow any more pressure, but he said, 'Young man, if I were you I would know how that table was derived.' I told him I would if I lived a week longer. I now have only four more days of grace."

"Well, Brown, you have brought up a big subject," said the Chief, "and although I can tell you how that table of spacing for stay-bolts was derived, I am not going to promise to tell you nearly all there is to the subject of stay-bolt spacing. In many cases the question of how to figure the direct tensile pull on each bolt is not a simple matter. In the case of rectangular spacing like this," continued the Chief, drawing Fig. 1, "the matter is simple, for by taking points midway between all the bolts surrounding the one to be considered, it will be seen that the surface to be supported by a single bolt will be that inclosed by the dotted lines, or an area equal to P times P'. Sometimes the arrangement is the same but turned at an angle with the axis of the boiler like this (Fig. 2), which often confuses the inspector, and he wants to call the distance indicated as a the pitch of the stays, but you can see that by turning the sketch so that line BB is the base, there is no real difference in the arrangement, as shown in Figs. 1 and 2.

"Now, while I say that the area supported by a single bolt is that inclosed in the dotted lines, that statement is not strictly true, because the area occupied by the least cross-sectional area of the bolt in the space between the stayed surfaces would have to be subtracted from this area to be supported, since no pressure can act on such surface to tend to separate the sheets that are stayed. However, in calculating the stress on stay-bolts, the area occupied by the bolt is usually not considered.

"When you come to a stay-bolt layout like this," continued the Chief, drawing Fig. 3, "you will have to sit up and take notice if you hope to tell just what part of the load each bolt carries. You will see that the arrangement of bolts here forms regular hexagons and that one-third of each bolt is concerned with the support of each of the hexagons. The problem is, how much area does each bolt support? Since there is one-third each, of six bolts concerned in the support of an entire hexagon as given in Fig. 4, two bolts are therefore provided for the support of the surface bounded by the six bolts, or one bolt supports an area of $1.3\ P^2$, since the entire area of a hexagon is equal to 2.6 times the square of one of its sides. The distance P is also the pitch that determines the pressure that can be allowed on the stay-bolted surface on account of the thickness of the plate, as I will show you presently.

"Now you can see", continued the Chief, "that even with stay-bolts symmetrically spaced, it may be difficult to say just how much area is supported by a single bolt and with unsymmetrical spacing it is practically impossible to determine the amount of load carried by each bolt. As to the main question, how the maximum spacing between the stay-bolts is determined according to the thickness of the sheet and the pressure required, that is difficult to answer, for this problem has never been solved in an entirely satisfactory manner. Unwin states that if a flat plate is supported by stay-bolts arranged in lines and with the bolts uniformly spaced each way, as was shown in Figs. 1 and 2, $P = \frac{4.5\,ft^2}{a^2}$, expresses the relation between the greatest stress in the plate and the pressure. Here a is the pitch of the bolts in inches, t the thickness of the plate in inches, P the pressure in pounds per square inch, and f the maximum fiber stress in the plate.

"Now while Unwin's formula no doubt expresses the relation between these factors properly, it is not used for the purpose of obtaining the maximum spacing of stay-bolts, and probably because experience has demonstrated that when the sheet between the stays deforms too much, there is a tendency for the sheet to unbutton, that is, to pull off the ends of the stay-bolts. The usual method of arriving at the maximum spacing between stay-bolts is by assuming that the sheet is supported by the bolts

in the same manner as it would be if there were beams placed below the rows of bolts in one direction, like this," and the Chief drew Fig. 5, "and that the plate between these supporting beams acts like a beam that is fixed at the ends and uniformly loaded.

"Now, when the total uniform load on a beam of this character is represented by W, f represents the maximum fiber stress as before, b and d represent the breadth and depth of the beam respectively, and L indicates the length between supports, the formula representing the relation between these factors is $W = \frac{2fbd^2}{L}$.

"You can find this in any work giving the strength of beam," continued the Chief, "for it is merely the formula for a uniformly loaded beam having fixed ends. Now, to get it into terms to suit our purpose, we will consider a strip of the stayed sheet 1 in. wide and let P represent the pressure of steam in pounds per square inch; then W would become P times the pitch, or the distance L in Fig. 6, and b we have already assumed as equal to one, and d^2 would become t^2, where t represented the thickness of the plate. So our formula, $W = \frac{2fbd^2}{L}$, would become $PL = \frac{2ft^2}{L}$, or $P = \frac{2ft^2}{L^2}$.

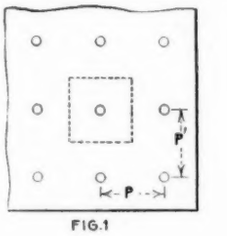


FIG.1

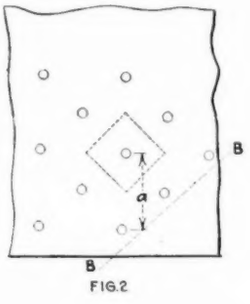


FIG.2

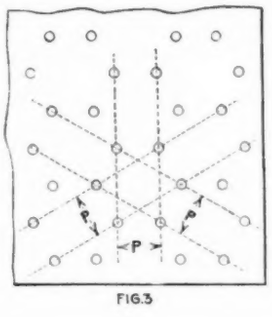


FIG.3

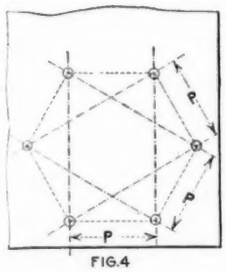


FIG.4

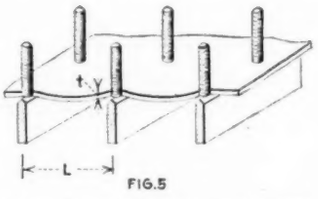


FIG.5

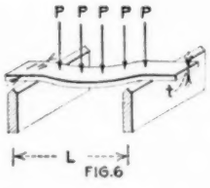


FIG.6

VARIOUS STAY-BOLT SPACING

"This is the formula that is used in the Lloyd's rules, and it has been widely copied by many other authorities on the subject of boiler construction. In the Lloyd's rules, t is given in sixteenths of an inch instead of in inches direct, and since t is squared, the Lloyd's formula would read $P = \frac{2f\frac{t^2}{(16)^2}}{L^2}$ or $P = \frac{\frac{2f}{256}t^2}{L^2}$ but the expression $\frac{2f}{256}$ in Lloyd's formula is specified as a constant C and it is given various values according to the circumstances of the case. For example, with plates not over $\frac{7}{16}$ in. in thickness and where the stays are merely screwed through the plate and the ends riveted over, the value of C is taken at 90. Therefore $\frac{2f}{256} = 90$ or $f = 11{,}520$ lb. which you will admit is a conservative working fiber stress for steel plate. For plates over $\frac{7}{16}$ in. thick, Lloyd's formula gives the value of C as 100, which corresponds to working stress of 12,800 lb. and for stays fitted with nuts and washers still higher values for C are given, but these do not necessarily represent an increase in the maximum fiber stress of the plate, since the effect of such fastenings is to distribute the load over the sheet in such a way as to diminish the maximum fiber stress that would be expected owing to the span.

"Now, Brown," continued the Chief, "the table of maximum spacing for stay-bolts is, as you see, based on the strength of the sheet between the bolts when figured as a beam and would indicate that the maximum stress in the plate was at a point midway between the stay-bolts; yet we know from experience that stayed sheets do not fail in that way, and experiments on stayed surfaces seem to indicate that the maximum stress in the sheets comes as a tension on the inside of the sheet, on the part immediately surrounding the stay-bolt. You can see how it is that the matter of the proper spacing for stay-bolts is not very satisfactorily concluded, but it may be said to be almost as satisfactory as our method of estimating the strength of the boiler shell on the calculated strength of the net section of the longitudinal joint, when, as a matter of fact, this section of the sheet almost never fails when an explosion occurs.

"The British Board of Trade rule for the spacing of stay-bolts over flat surface is $P = \frac{C(t+1)^2}{S^2 - 6}$ for stays spaced at equal distances each way, where P and t have the same significance as before and s equals L or the pitch between the rows of bolts. You can see that this is not a beam formula, although it gives practically the same results as the Lloyd's formula up to the usual maximum pitch of screwed stay-bolts, except for high pressures and thin plates, where the Board of Trade formula gives wider spacing.

"As I have already told you," continued the Chief, "in the case of Figs. 3 and 4, the distance P represents the pitch of bolts that should be used to determine the pressure that could be allowed on the stayed sheet, owing to its thickness. You can see that this is correct, because the dimension P is the greatest distance between adjacent rows of bolts, and therefore, when used in place of L in the formula, would give the least value for the pressure to be allowed."

Just then the train whistled for the flag station to which the Chief and Brown were bound, and getting off, they started on their trudge to the sawmill.

A Good Lubricant must have enough body to keep the surfaces between which it is introduced apart; a low coefficient of friction in actual use; high capacity for carrying away heat; freedom from tendency to decompose or to gum, either before use or during use; absence of acids or other substances which would act upon the bearing surfaces; high temperatures of evaporation and decomposition; low temperature of thickening; freedom from grit or other foreign substances, and last, special adaptability to the work to be done. Thus a lubricant may be very good for light bearings, but very poor for heavy ones and vice versa.

Panama-Pacific Exposition--X

The Chas. A. Schieren Co., of New York, not content with the usual exhibition of rolls of belting, has installed an operative exhibit. It consists, as shown in Fig. 3, of a booth octagonal in form, the structural-steel columns of which, in addition to supporting the dome with a 22-in. double-ply leather belt 150 ft. long, carry the hangers of three lines of shafting used to illustrate fourteen different drives, quarter turns, fifth turns, mule drives, horizontal and vertical, etc.

Four Lenix drives are used to solve four different belting problems. Upon one of the extended pedestals at the base of the column is a centrifugal machine, the belt of which travels 5,700 ft. per min., making a full half-turn. On another pedestal a cement-tube-mill grinder is run at the rate of 60 r.p.m. by means of a simple countershaft running at three times that speed. On the third pedestal is a quarter-turn mule drive used in silk mills, with a tachometer; and on the fourth, a right-angle mule drive, one belt running over four pulleys each at right angles to the next.

Belt Runs Ten Months Under Water

In the center is a fountain in a 6½-ft. basin in which is a Lenix drive operated by a vertical shaft; and a duck, the trademark for this firm's Duxbak belting, performs antics upon the surface of the pond, due to his connection with the belt, which has run under water ever since the exposition opened.

The Graton & Knight Manufacturing Co.'s exhibit comprises three features: A complete line of their belting arranged for easy examination, shown in Fig. 1 on page 785, a working exhibit, shown in Fig. 2; and a demonstration of the qualities of their Spartan belting. Two of these belts drive through a chamber heated to 200 deg. and from there into a bath of oil and water. The details of the working exhibit are indicated by reference to the numbered items of which it is composed.

The Dodge Manufacturing Co., of Mishawaka, Ind., has in the Herron, Rickard & McCone space an exhibit, of which a particular feature is a wood-rimmed pulley that in actual test has been run at 2,400 revolutions, attaining a rim speed of 28,889 ft., or 5⅜ mi. per min.

A rope drive upon the one-rope, or American, system is shown in operation with an automatic electric cutoff for releasing the clutch when a rope breaks.

Running Chain Appears to Stand Still

The Link-Belt Co., of Philadelphia, has an arrangement of silent chain drives in the exhibit of Meese & Gottfried Co., by which the observer, looking through the revolving slotted disk synchronized with the moving chains, sees the chains and their pinions apparently stationary or moving very slowly.

The same exhibit contains a number of steel, wooden, groove-rimmed and other pulleys, hangers and transmission machinery, and a 12-ft. double-armed steel pulley revolving in open bearings with no apparent source of power.

Of the various systems for removing the scale-forming materials from boiler-feed water the only one on exhibition is that of the Permutit Co.

The user of hard water has probably often heard in the last few years references to "permutit" and "artificial zeolites," but has figured that no water could be harder than those terms, and so passed the matter up. However, as like cures like, some knowledge of the subject may come in handy.

Silicates are, as is probably known, combinations of silica, which is seen in almost pure form as quartz, and the oxide of a metal. Ordinarily silicates are insoluble in acids (except hydrofluoric)—as is, for example, glass, a very common silicate. Zeolites, on the contrary, are silicates soluble in acids and contain water and alumina, besides another metallic oxide. As another and vital distinguishing characteristic, the groups of metallic oxides other than alumina are attached to the silicate group through the alumina group. This, however, while an interesting chemical fact, does not enter into the working of artificial zeolites as water softeners.

The manufacture of permutit consists in fusing ordinary feldspar (which contains sodium or potassium, aluminum and silica), kaolin (containing alumina and silica), pearlash (crude potassium carbonate) and soda. This forms a sodium- or potassium-aluminum silicate, which is then treated with hot water, which dissolves the excess of the alkali (soda or potash) and adds the water necessary for the formation of a zeolite. The sodium permutit is, in moist condition, of a granular or flaky form with a luster like that of mother of pearl, and is exceedingly porous.

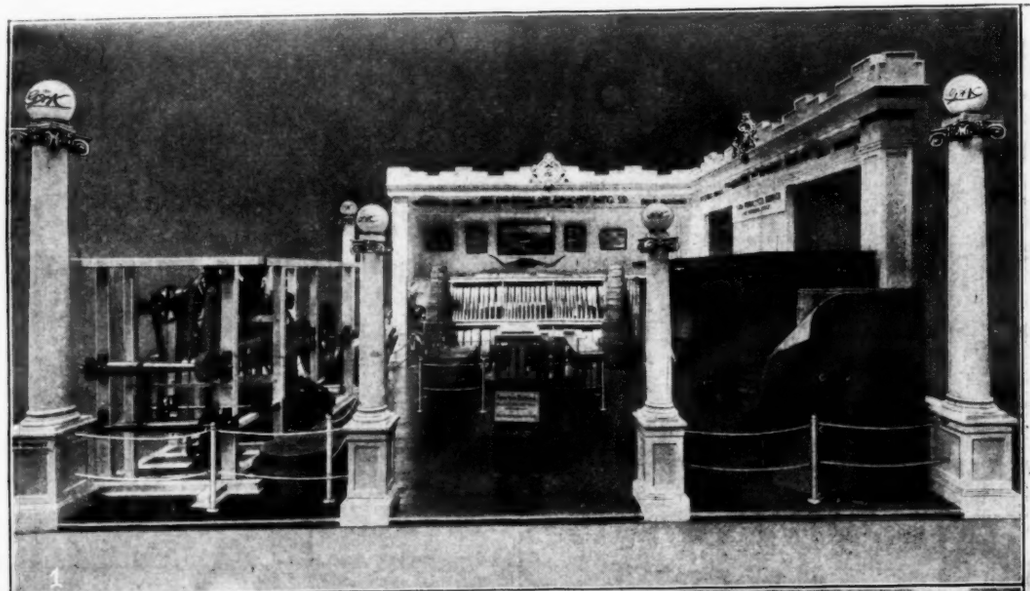
How Permutit Absorbs Scaleforming Salts

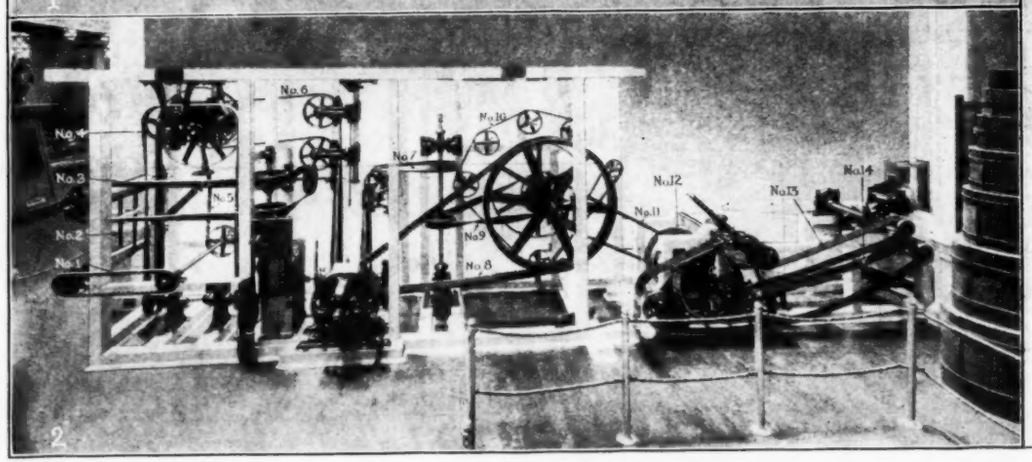
The silicate formed in this way is extremely unstable, and if solutions of calcium or magnesium salts not containing large amounts of soda or potassium as compared to the calcium or magnesium are passed over it, the soda or potassium is replaced in the permutit by the calcium or magnesium, and the sodium or potassium remains in solution. Hence, either temporary hardness (calcium or magnesium bicarbonate) or permanent hardness (calcium or magnesium sulphate) is removed in this way, with formation of sodium bicarbonate or sulphate, which remain dissolved in the treated water, but do not form boiler scale.

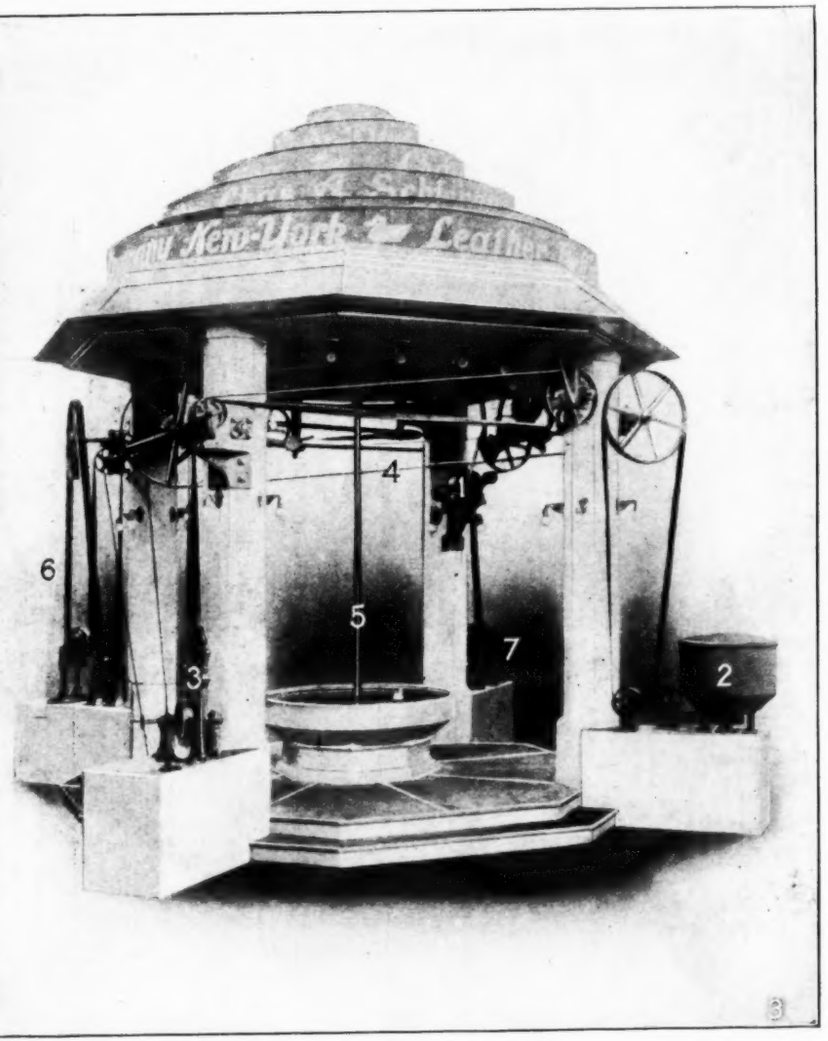
In use, an upward-flow filter is filled with permutit, and a layer of gravel between perforated plates placed above to prevent loss of the permutit by entrainment in the outflowing water. The upward-flow filter is stated to be more effective than the downward-, probably because the material naturally tends to cake together, which tendency is assisted by a down flow and retarded by the loosening action of an upward current.

The more slowly the water passes through the permutit mass, the more complete the elimination of hardness. In practice a velocity of 10 to 15 ft. per hr. has been found most suitable for obtaining good results with comparatively thin layers. But the thickness of the layer and the rate of filtration must be determined from analysis of the water to be treated.

So much for the removal of the scale-forming elements from the water, until the point is reached that all the available soda of the permutit is replaced by calcium and mag-







SOME OF THE EXHIBITS OF LEATHER BELTING IN THE PALACE OF MACHINERY AT THE PANAMA-PACIFIC EXPOSITION

Panel 1 is a general view of the exhibit of the Graton & Knight Manufacturing Co., of Worcester, Mass., and Panel 2 a front view of the working exhibit seen endwise at the left in the upper panel This exhibit comprises drives of various sorts made with their "Neptune" and "Spartan" belting. The various drives, which may be identified by the numbered items upon the engraving are as follows:

1. Single-strand V-belt drive ⅞ inch.
2. Cone drive (textile machinery).
3. Mule drive in horizontal plane.
4. Bicycle drive (cotton mill spring frames).
5. Vertical drive with weighted idler.
6. Quarter-turn drive with two idlers.
7. Mule drive in vertical plane.
8. Five-strand 1¼-inch V-belt drive 7: 1.
9. Quarter-turn drive without idler.
10. Stripper drive (woolen card).
11. Single-strand V-belt drive 1½ inch.
12. Matcher head drive. } High-speed
13. Upper cylinder drive. } planer and
14. Lower cylinder drive. } matcher.

Panel 3 is the exhibit of Chas. A. Schieren Co., of New York. The roof supports a cone of belting; upon the pillars are mounted hangers, upon the shafting carried in which are demonstrated drives of various sorts. A "Duxbak" belt has been running submerged in the water of the central basin since the exposition started.

1. Motor connected by "Lenix" drive to shaft No. 1. Speeds, 1,300: 360.
2. Centrifugal machine driven at 1,080 r.p.m. through "Bamag" counter.
3. Quarter-turn mule drive for silk mills, with tachometer.
4. "Duxbak" belt with tension pulley driving lineshaft 2.
5. Central shaft running at 3 r.p.m. and driving "Duxbak Lenix" under water.
6. Right-angled drive between shafts 2 and 3.
7. Cement tube-mill grinder driven by crossed belt and "Lenix."

nesium. But if one must buy new permutit each time that the available sodium of the zeolite is exhausted, the process would be too expensive. Fortunately the calcium

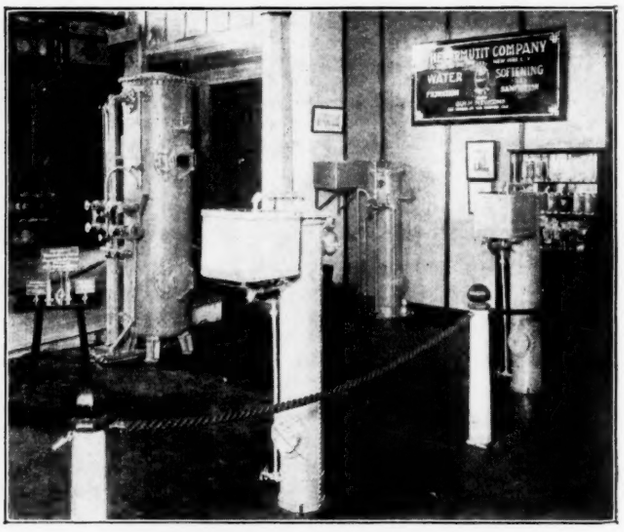


FIG. 4. EXHIBIT OF THE PERMUTIT CO.

permutit is also unstable, and just as a soluble calcium salt decomposes the sodium permutit, so will a sodium salt decompose the calcium permutit. That is, the reaction is a reversible one. But while dilute solutions of calcium affect the sodium permutit, a concentrated sodium solution is needed to decompose the calcium permutit.

Sodium chloride (common salt) is cheap, so a 10-per cent. sodium-chloride solution is used to regenerate the exhausted permutit. All water is drawn from the filter and the salt solution allowed to flow for eight or ten hours. The excess salt water must then be washed out of the filter with clear water, about 300 gal. of clear water usually being necessary per 100 lb. of salt used, and the cycle of operations resumed.

It may be further noted that manganese permutit will remove iron and manganese from water, except the iron be present as an organic salt, in which case preliminary treatment with alum is necessary; but the chemistry of the process is rather complicated, and as iron removal is usually the problem of a laundry or dye works and not of a power plant, it seems sufficient to state the fact without giving the full chemical explanation. The exhibit of the Permutit Co. is shown in Fig. 4.

Cooper's Glue, for Joints in Wooden Tanks, is made of a mixture of boiled linseed oil with double its bulk of good glue thoroughly stirred and applied at about 200 deg. F. to the joints during construction.

Standard Pressure Valves and Fittings are intended for a working pressure up to 125 lb. per sq.in. Medium pressure means for a pressure of 125 to 175 lb. per sq.in. For higher pressures the manufacturer should be consulted and special metals used.

Bends for Pipe-Line Expansion

By W. L. Durand

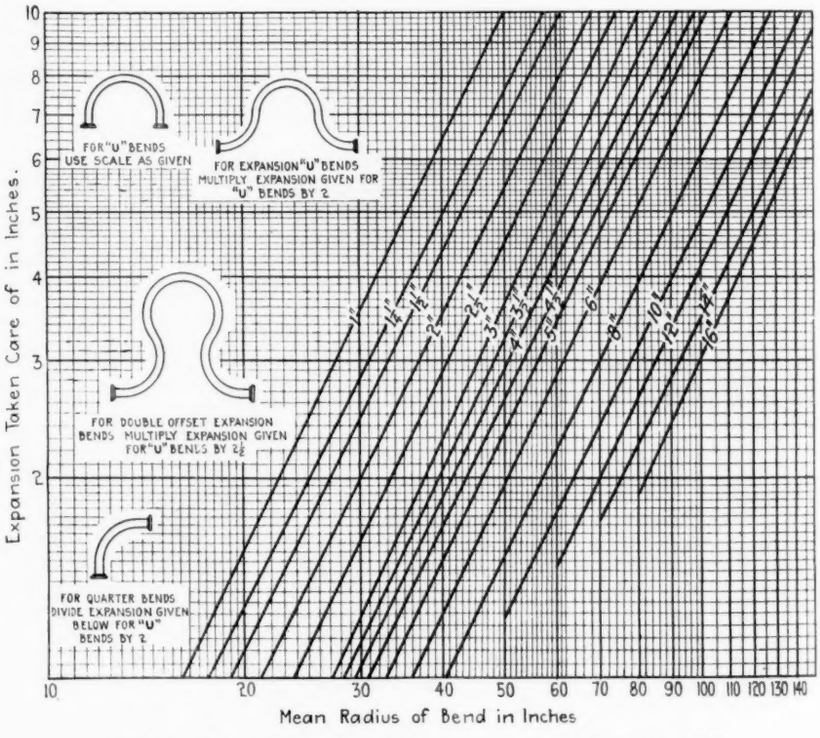
In recent years the use of pipe bends in place of expansion joints has become so common that the shapes have become standardized in form. They have the advantage that once in place they require no attention, whereas expansion joints are often a source of trouble.

While it is an accepted fact that the expansion value of a bend is inversely as the thickness of the wall and directly as the radius, no exact information has been available as to how much expansion any given bend will take care of without damage to itself or to the adjacent piping. To supply this information the Crane Co. has conducted an extensive series of tests covering numerous types and sizes of pipes and published the results in the form of curves on standard cross-section paper.

The curves radiate from a common center, and consequently the lines are so close together for the smaller values that it is difficult to read them. If plotted on logarithmic cross-section paper, the curves become straight lines with all the advantages of this form, so I have replotted them and added those for 1¼- and 1½-in. (which are not given in the original) calculated as follows:

The formula for U-bends is

$$E = \frac{0.0052\ R^2}{d}$$



EXPANSION ANY BEND WILL ALLOW WITHOUT INJURY

in which

E = Expansion in inches;
R = Mean radius of bend in inches;
d = Outside diameter of pipe in inches.

If any two of the three values are given, the third can be easily found from the curves. Example: What is the necessary radius for a U-bend to take care of 3 in. of expansion in an 8-in. pipe? Referring to the curves and running out horizontally from 3 in. to the line marked 8 in., the radius of the bend is read as 18 in.

Some Recent Experiments in Surface Combustion--I

By L. J. Bradford* and C. D. Corwin*

SYNOPSIS—This, the first of two articles on the subject by the authors, is a summary of the literature on surface combustion, telling briefly of the work of Bone and Lucke.

The phenomenon of surface combustion was first observed by Davy in 1823 and discussed by Graham, Faraday and Dulong. No satisfactory explanation was reached, and the matter was dropped with no practical results attained.

There are two kinds of combustion—homogeneous, taking place throughout the system as a whole, and heterogeneous, occurring in layers in contact with an incandescent surface. The latter is the more rapid and is the kind present in surface combustion.

It has been found that all hot surfaces accelerate combustion, the extent of their action being dependent on the temperature and the character of the surfaces. The surfaces should be incandescent. Surface combustion itself is accelerated by first putting the surface in contact with the gas to be burned. If it is brought into contact with oxygen first, the combustion is retarded.

A practical system of surface combustion has been devised by Wilson and Mathiesons, Leeds, England, but it does not seem to have been much used. There seems to be no good reason for its use not being more general, as the meager literature published on the subject mentions many advantages and few disadvantages.

Catalysis Aids Combustion

In his experiments Sir Humphrey Davy noticed that a mixture of two gases that would ordinarily unite only at a high temperature could be made to combine at a much lower initial temperature by passing them through platinum sponge. He also found that if a mixture of oxygen and hydrogen was passed over finely divided platinum or through a platinum sponge, it would ignite, even though both gases and the platinum were initially at room temperature. The platinum did not take part in the chemical reaction, but merely caused the two gases to combine at a temperature below that at which they would normally do so. This action is known as catalysis. No satisfactory explanation of catalysis has ever been given, though there seems to be reason to believe that it is due to the catalytic agent ionizing the atoms of the gas adjacent to its surface and thus hastening combustion. It has also been found that all surfaces act more or less as catalytic agents and that their power increases with their temperature. For example, combustion is materially assisted when a mixture of gas and air is forced through an incandescent porous substance. The combustion becomes flameless and takes place much more rapidly than it would under ordinary conditions. Apparently, all combustion occurs on the surfaces of the granules composing the porous material, and hence the name "surface combustion" has come to be applied to this form of combustion. It is characterized by an absence of flame, high temperatures and remarkable efficiency, complete combustion being obtainable with practically no excess of air.

It is only within the last decade that surface combustion has been applied to industrial purposes, and as far as can be ascertained, only two men have conducted any work looking toward its utilization; namely, C. E. Lucke in America and W. A. Bone in England.

Bone's efforts have been directed principally to applying surface combustion to industrial apparatus, notably boilers and crucible furnaces. His first device consisted of a diaphragm, Fig. 1; a mixture of combustible and air was introduced at *A* and passed through a plate of porous firebrick which formed the face of the diaphragm. It was found that all the burning took place in the outer ¼ in. of the plate, that it was flameless, and that when using coal gas a temperature of 800 deg. C. (1,472 deg. F.) could be obtained. Fig. 2 shows a typical crucible furnace using surface combustion. With such a furnace Bone attained a temperature of 1,880 deg. C. (3,416 deg. F.).

Fig. 3 shows an experimental boiler to which Bone had adapted surface combustion. The tubes were plugged at

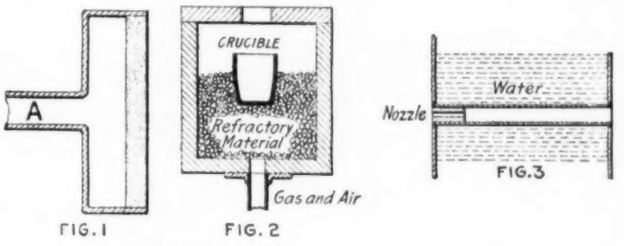


Fig. 1—Bone's diaphragm for surface combustion. Fig. 2—Section of surface combustion furnace. Fig. 3—Section of part of boiler where surface combustion occurs in the tube

one end with a fireclay nozzle having a ¾-in. hole for the introduction of the mixture of gas and air. The remainder of the tube was filled with broken firebrick. Gas and air were supplied in quantities theoretically correct for complete combustion, and under a pressure of 17.3 in. of water. 99.6 cu.ft. of gas at 760 mm. (30 in.) Hg and 15 deg. C (50 deg. F.) was used per hour. The boiler was lagged so as to make the radiation less than 2 per cent. Under these conditions he obtained an efficiency of 94.3 per cent.

From available published matter it appears that Lucke has devoted his experiments chiefly to applying surface combustion to domestic uses. Practically all his work has been done with gas and air at low pressures, seldom exceeding 1 in. of water. The rates of combustion have also been comparatively low. He investigated various refractories and found that white alundum was the best for withstanding the high temperatures obtained.

Both authorities agree that through the use of surface combustion gas may be burned completely with practically no excess of air and that the resulting temperatures are high: Bone placing the upper limit with natural, water or coal gas at 2,500 deg. C. (4,532 deg. F.).

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From the writings of these men it appears that the greatest difficulties encountered are those that arise from backfiring, and fusing of the refractories. Since the gas and air are mixed in proportions theoretically necessary for complete combustion, it is clear that unless steps are taken to prevent it the flame will run back down the supply pipe and perhaps cause disastrous explosions. This can be prevented in two ways—first, by chilling the flame by means of some form of screen in the pipe and thus extinguishing it, and secondly, by supplying the mixture at a velocity greater than the velocity of flame propagation.

Of the two methods the latter is the better, for unless special precautions are taken the screen will eventually become hot and no longer hold back the flame, while the second method is effective even though those portions of the passages adjacent to the fire become red-hot. The velocity should be above 5 ft. per sec.

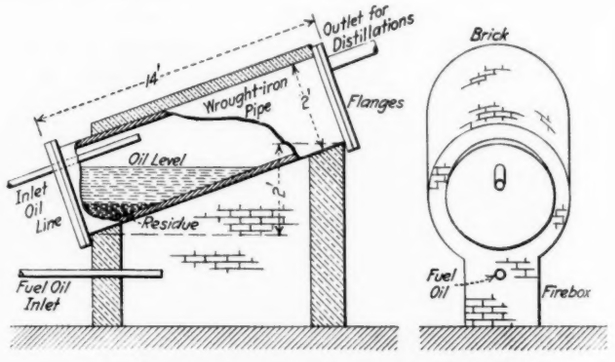
Operating a Diesel Engine Under Difficulties

The incident of the dredging company in Alaska, recounted by J. H. Niles, indicates clearly that resourcefulness has not vanished as an attribute of the American engineer and that pioneering is still going on all about us as interesting as that of fifty years ago.

A 200-hp. 4-cylinder Diesel engine was purchased by a dredging company in the Seward Peninsula in Alaska. It was shipped by boat and arrived early in the spring of 1913. The erecting engineer that accompanied the shipment was taken suddenly ill and died on the ship, with the result that the master mechanic of the dredging company had to unload the machinery from the boat and supervise its erection. The unloading was comparatively simple, and the erection of the engine (which was continued day and night, as in the summer in Alaska there is no time when it is not perfectly light) was completed in 70 working hours. When the plant was ready to start it was found that the high-pressure air of 1,000 lb. had escaped from the starting tanks, and it was impossible to start the engine without recharging these tanks. A traction engine was therefore belted to the flywheel of the engine and turned the machine over at approximately 80 r.p.m., whereas its normal speed was 200 r.p.m. In this way in 27 hours the air pressure was replenished in the starting tanks and the engine was ready to be turned over. It was then found that the direct-connected air compressor would not supply sufficient pressure to operate the engine under load, the pressure being only sufficient to inject the fuel at practically no load. It was therefore necessary to change the clearance in the lower stages of the compressor so that it would give a higher final pressure, but a considerably less volume of air. During these trials the outlet valves in the high-pressure stage dropped into the cylinder and the cylinder head was destroyed. It was necessary to go to Nome, 80 miles away, and have a new head cast in a foundry equipped for making grate bars and car wheels. By eliminating all the cores a good casting was finally made, which worked satisfactorily for the entire season.

After the air compressor was in perfect operating condition, it was found that the oil upon which the engine was intended to operate was of such a low gravity (14 deg. Bé.) and had such a large percentage of asphalt (above 50 per cent.) that it would not properly burn in the engine for any length of time. The company had 50,000 bbl. of oil in storage, and it was impossible to get a new supply in time for the season's operation, which was limited to about 80 days. It was therefore necessary to again call upon the resourcefulness of the master mechanic, who tried distilling a portion of the oil in the laboratory, and the crude manner in which this was accomplished can be gained by the fact that there was no thermometer available to test the temperature at which the distillate was going over. After refining about 10 gallons of oil in the laboratory and finding that it operated satisfactorily, a crude retort was made somewhat in the form shown in the accompanying sketch. This retort produced 1,600 lb. of oil in 24 hr. and 200 lb. of residue, which was scraped out after each run.

The cost of the oil after it had been treated was 13c. per gal.—8c. per gal. original cost and 5c. for treatment—as against 18c. for lighter-gravity oil of 20 to



AN IMPROVISED REFINERY IN ALASKA

24 gravity delivered at Seward Peninsula. This method of treating the oil was continued throughout the season, the engine operating on it entirely satisfactorily. In order, however, to take advantage of every day's operation possible, the entire crew of the dredge remained after the season (that is, water navigation) had closed, and it was necessary for them to go overland 1,000 mi. by dog team. All of the men had to walk the first 500 mi., the rest of the distance it was possible for them at some time to ride upon the sledges. During one day but 4 mi. was made, although an average of about 35 mi. was made throughout the trip.

The experience of this company with the engine shows conclusively that, particularly in shipments out of the United States, extreme care should be exercised by manufacturers in making their apparatus so that such difficulties as encountered by this company would be almost impossible; and the assurance to customers of the operation of the engine on any kind of fuel oil, when such is not practicable, reflects upon the Diesel industry as well as upon the manufacturer making such guarantees.

Necessary Qualities in Oils vary to some extent, but first of all always, there must be sufficient body, viscosity, for the pressure in order to keep the surfaces from harmful contact, that is to do the work. The other qualities may rank differently in importance under differing conditions. They are freedom from corrosive acids, fluidity, minimum coefficient of friction, high flashpoint (for safety) and freedom from oxidation or "gumming."

Fundamental Principles of Alternating Current--I

By F. A. Annett

SYNOPSIS—A simple explanation of the sine curve as commonly used in alternating-current work; to be followed by a discussion of the effective e.m.f. in an alternating-current circuit.

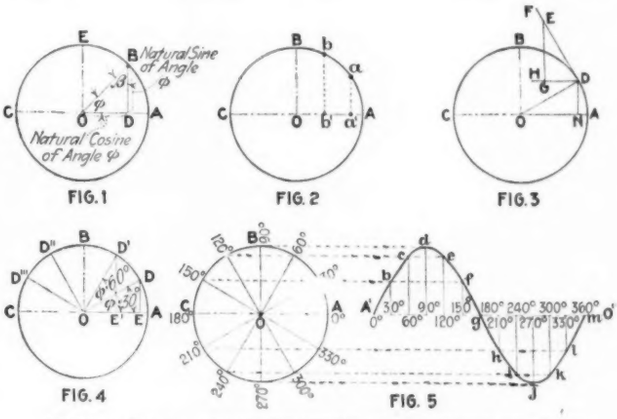
For a thorough understanding of alternating current one must first get a clear conception of the underlying principles, which will here be taken up briefly.

An angle may be considered as generated by the radius of a circle revolving about a point at the center of the circle; thus, in Fig. 1, if the radius OB be revolved counter-clockwise about the point O from the horizontal position OA to the position OB, angle BOA will be generated, and if a perpendicular is let fall from B to the opposite side OA, the right triangle OBD will be formed. In any right triangle the sine of either acute angle is the ratio of the opposite side to the hypotenuse; thus, in Fig. 1, the sine of angle ϕ equals $\frac{BD}{OB}$, or the sine of angle $\beta = \frac{OD}{OB}$. If side OB is given a value of unity since ϕ will equal $\frac{BD}{1} = BD$. When OB coincides with OA, angle ϕ will equal zero and BD will also equal zero; therefore the sine of angle ϕ will be $\frac{O}{OB} = O$. When OB has revolved up into the vertical position OE, BD will be equal to and coincide with OB; therefore, sine 90 deg. will be $\frac{OB}{BD} = \frac{1}{1} = 1$. Thus it is seen that the sines of angles between zero and 90 deg. vary from zero to unity; these values are given in Table 1.

The cosine of an angle is the ratio of the adjacent side to the hypotenuse; thus the cosine of angle $\phi = \frac{OD}{OB}$. When angle ϕ is zero, OB coincides with and is equal to the side OD; therefore, cosine 0 deg. $= \frac{}{OB} = \frac{1}{1} = 1$. As OB revolves around to the vertical position OE, OD decreases in length until, when OB coincides with OE forming an angle of 90 deg. with OA, OD becomes zero; therefore cosine 90 deg. $= \frac{OD}{OB} = \frac{O}{1} = O$. From this it will be seen that the cosine of the angle decreases from unity for an angle of 0 deg. to zero for 90 deg., which is just the opposite to the sine. The values given in Table 1 for the sines and cosines are the natural functions and should not be confused with the logarithmic values.

Distance Traveled and Rate of Motion

One must be careful to distinguish between the distance through which a body moves and its rate of motion. A train in going from New York to Philadelphia travels approximately 90 mi., and if the run is made in 120 min., the average rate of motion would be ¾ mi. per min. In this case 90 mi. is the distance traveled and ¾ mi. per min. the average rate of motion. If for each minute that the train is in motion it travels ¾ mi., it will be traveling at a uniform rate and may be said to have uniform motion; that is, it will move through equal units of space in equal units of time. For this to be possible the train would have to be moving at ¾ mi. per min. at the beginning and at the end of the run. If it were to start from rest and travel the 90 mi. and come to rest again in 120 min., although the average rate of motion would be ¾ mi. per min., the train would have a varying motion, for it cannot start and stop instantaneously, but must start from rest and gradually increase to full speed, which must be greater than ¾ mi. per min. to make up for lost time at starting and stopping and gradually coming to rest at the end of the run. The rate of increase



FIGS. 1 TO 5. ILLUSTRATING HARMONIC MOTION

in motion is called acceleration, whereas the rate of decrease is called deceleration, or retardation. There are several kinds of varying motion, but the one that is most important in the study of alternating current is harmonic motion.

Explanation of Harmonic Motion

In Fig. 2 suppose a body to be moving around the semicircle ABC at a uniform rate and also a second body moving across the diameter AC at such a rate that it will always remain in the same vertical plane as the body moving around the path ABC; that is, when the body moving around the semicircle is at a the body moving along the diameter will be at a', as indicated, or when the first body is at b the second will be at b', etc. From this it will be seen that although the body moving around the semicircle has a uniform motion, that moving across the diameter has a varying motion starting from rest at A, reaching a maximum at O, and then decreasing in speed until it comes to rest again at C. This body is said to have harmonic motion and will be referred to later.

When the two bodies are in the vertical plane BO, Fig. 2, they are both moving in the same direction and at the same rate. If the radius of the circle ABC in Fig. 3 is let to represent the maximum rate of motion of the body moving along the diameter AC and consequently the rate of motion of the body moving around the semi-

circle *ABC*, the rate of motion of the former at any point along the diameter is equal to the perpendicular distance (such as *DN*) between the two bodies at any instant. It will be observed that *DN* increases in value from zero at *A* until it is equal to the radius *BO* at *B*. Following is a proof of this rule:

The direction of motion at any instant of a body moving in a circle is always tangent to the circumference, or at right angles to the radius. In Fig. 3 consider the instant when the body is at *D*, at which time it will be moving at right angles to the radius *DO*. Draw the line *DF* at right angles to *DO*; then *DF* represents the direction of motion at this instant. On *DF* lay off the distance *DE* equal to the radius *DO* (equal to the rate of motion of the body moving around the circle). Draw *DH* parallel to the diameter *AC* and from *E* let fall a perpendicular to *DH*, thus forming the right triangle *DEG*. Now if *DE* equals the rate of motion of the body around the circle, *DG* will equal the rate of motion at this instant of the body moving along the diameter *AC*. The right triangles *DEG* and *DON* are similar (their corresponding sides being perpendicular); therefore *DE* : *OD* = *GD* : *DN*. But as *DE* was made equal to *OD*, it follows that *GD* must equal *DN*. Since *DG* = the rate of motion of the body moving along the diameter, *DN* the perpendicular distance between the two bodies at any instant will also equal the rate of motion of the body along the diameter.

To illustrate, assume in Fig. 4 that the body traveling around the semicircle *ABC* is moving at the rate of 15 ft. per sec., which is also the maximum rate along the diameter *AC*. Let the radius *OA* represent this rate, and consider the instant when the body has moved through 30 deg. on the semicircle, as shown by the position *D*. The perpendicular *DE* will represent the rate along the diameter *AC* at this instant. By joining *O* and *D* the triangle *ODE* will be formed. Then sine 30 deg. $= \frac{DE}{OD}$, whence *DE* = *OD* sine 30 deg. In this case *OD* = 15 ft. per sec. and sine 30 deg. = 0.5 (see Table 1). Therefore, *DE* = 15 × 0.5 = 7.5 ft. per sec., which is the rate at which the body along the diameter is moving at this instant. This must not be confused with the distance traveled, as the body has moved only the distance *AE*, which in this case is about 2.1 ft., but in moving this distance the body has been accelerated to about 7.5 ft. per sec. At 60 deg. *D'E'* = *OD'* sine 60 deg. = 15 × 0.866 = 12.99 ft. per sec., and the body has moved the distance *AE'* or 7.5 ft. In each case the rate of the body moving along the diameter *AC* is proportional to the sine of the angle through which the body traveling around the circle has moved. On the line *A'O'* Fig. 5, lay off distances proportional to the divisions of the circumference of the circle *ABC* in Fig. 4, and project horizontally the various points on the circle to meet the vertical ordinates erected at corresponding points on the line *A'O'* thus locating points *A'*, *b*, *c*, etc., respectively. By joining these points a curve will be formed as shown. The vertical distance from any point in this curve to the base line *AO* will be proportional to the sine of the corresponding angle in the circle *ABC*. For this reason the curve thus formed is called a sine curve. This is the curve of electromotive force generated by a conductor revolving about an axis at a constant speed in a uniform magnetic field, as will be shown later.

Principles of the Dynamo-Electric Machine

When a conductor is moved in a magnetic field so as to cut the lines of force there is always a difference of potential developed between its terminals, or in other words an electromotive force is induced in the conductor. This is the fundamental principle of all direct- and alternating-current generators. The direction of the electromotive force depends upon the direction of motion of the conductor and the direction of the lines of force in the magnetic field. The magnitude of the e.m.f. is directly proportional to the number of lines of force cut per unit of time; this for a given uniform magnetic field depends upon the rate at which the conductor is moving and the angle it is moving through, with reference to the lines of force. In Fig. 6 let the line *ab* represent the direction and rate of motion of the conductor, when it is moving at right angles to the lines of force (across the magnetic field). When moving at a different angle, such as that shown at *a'b'*, the conductor will not cut

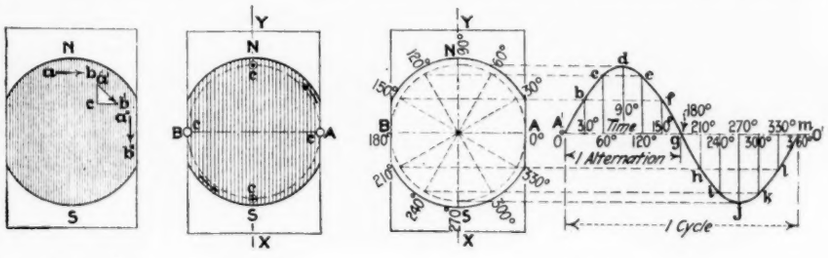


Fig. 6 Fig. 7 Fig. 8

FIGS. 6 TO 8. SHOWING APPLICATION OF SINE CURVE

so many lines of force in a given time, although it moves at the same rate. Here it has moved only a distance at right angles to the lines of force equal to the projection of the line *a'b'* on the horizontal as indicated by the line *cb'*, which represents the rate at which the lines of force are cut for this direction of motion through the magnetic field. When the conductor is moving parallel with the lines of force as represented by *a"b"*, it is not cutting any lines of force; consequently it is generating no e.m.f. From this it is obvious that the e.m.f. generated by a conductor moving at a constant rate in a uniform magnetic field depends upon the angle at which the conductor is moving with reference to the lines of force—varying from zero when the conductor is moving parallel with the lines of force to a maximum when it is moving at right angles to them.

Alternating Electromotive Force

When a conductor is revolved about an axis at a constant rate in a uniform magnetic field, its rate of motion is the same with reference to the neutral axis of the magnetic field as that of the body moving across the diameter *AC* in Fig. 2. In Fig. 7 the conductor *c* starts from the position *A* and revolves around in the dotted circle at a constant rate in the uniform magnetic field between the north and south poles. When it is in the neutral axis *AB* it is moving parallel with the lines of force; consequently, its motion with reference to the

horizontal axis of the magnetic field is zero; but as it moves around in its path, its motion across the horizontal axis increases until it is in the vertical axis *XY*, at which point it is moving parallel to the horizontal axis, and at right angles to the lines of force and is consequently generating the maximum e.m.f. Beyond this point the e.m.f. decreases until it again becomes zero when the neutral axis is reached at *B*. On the other half of the revolution the e.m.f. generated in the conductor will pass through the same series of values as on the first half, but in the opposite direction, for the conductor is moving across the magnetic field in the opposite direction. From the foregoing it will be seen that the motion of the conductor across the horizontal axis in Fig. 7 is the same as the motion of the body across the diameter of the semicircle in Fig. 2, or in other words, the conductor has harmonic motion across the magnetic field. Since the e.m.f. is directly proportional to the motion of the conductor across the magnetic field at any instant the electromotive forces generated at the different instances are sometimes spoken of as harmonic electromotive forces.

It was shown in Fig. 3 that, where the radius equaled the maximum rate, the rate of the body moving along the diameter at any instant is equal to the vertical distance between the two bodies for a corresponding period and proportional to the sine of the angle between the two bodies. If the radius about which the conductor is revolving in Fig. 7 is let equal the maximum rate at which the lines of force are cut, and therefore the maximum e.m.f., the vertical distance between the conductors and the horizontal axis at any instant will equal the rate at which the lines of force are cut, or the e.m.f. generated for a corresponding position.

TABLE 1. NATURAL SINES AND COSINES

Angle	Sine	Cosine	Angle	Sine	Cosine	Angle	Sine	Cosine	Angle	Sine	Cosine
0	0.0000	1.0000	23	0.3907	0.9205	46	0.7193	0.6947	69	0.9336	0.3584
1	0.0174	0.9998	24	0.4067	0.9136	47	0.7314	0.6820	70	0.9397	0.3420
2	0.0349	0.9994	25	0.4226	0.9063	48	0.7431	0.6691	71	0.9455	0.3256
3	0.0523	0.9986	26	0.4384	0.8988	49	0.7547	0.6561	72	0.9511	0.3090
4	0.0698	0.9976	27	0.4540	0.8910	50	0.7660	0.6428	73	0.9563	0.2924
5	0.0872	0.9962	28	0.4695	0.8830	51	0.7771	0.6293	74	0.9613	0.2756
6	0.1045	0.9945	29	0.4848	0.8746	52	0.7880	0.6157	75	0.9659	0.2588
7	0.1219	0.9926	30	0.5000	0.8660	53	0.7986	0.6018	76	0.9703	0.2419
8	0.1391	0.9903	31	0.5150	0.8572	54	0.8090	0.5878	77	0.9744	0.2250
9	0.1564	0.9877	32	0.5299	0.8481	55	0.8192	0.5736	78	0.9782	0.2079
10	0.1737	0.9848	33	0.5446	0.8387	56	0.8290	0.5592	79	0.9816	0.1908
11	0.1908	0.9816	34	0.5592	0.8290	57	0.8387	0.5446	80	0.9848	0.1737
12	0.2079	0.9782	35	0.5736	0.8192	58	0.8481	0.5299	81	0.9877	0.1564
13	0.2250	0.9744	36	0.5878	0.8090	59	0.8572	0.5150	82	0.9903	0.1392
14	0.2419	0.9703	37	0.6018	0.7986	60	0.8660	0.5000	83	0.9926	0.1219
15	0.2588	0.9659	38	0.6157	0.7880	61	0.8746	0.4848	84	0.9945	0.1045
16	0.2756	0.9613	39	0.6293	0.7771	62	0.8803	0.4695	85	0.9962	0.0872
17	0.2924	0.9563	40	0.6428	0.7660	63	0.8910	0.4540	86	0.9976	0.0698
18	0.3090	0.9511	41	0.6561	0.7547	64	0.8988	0.4384	87	0.9986	0.0523
19	0.3256	0.9456	42	0.6691	0.7431	65	0.9063	0.4226	88	0.9994	0.0349
20	0.3420	0.9397	43	0.6820	0.7314	66	0.9136	0.4067	89	0.9998	0.0174
21	0.3584	0.9336	44	0.6947	0.7193	67	0.9205	0.3907	90	1.0000	0.0000
22	0.3746	0.9272	45	0.7071	0.7071	68	0.9272	0.3746			

In Fig. 8 if several positions are taken as shown, to indicate the location of the conductor at different points in its revolution a curve may be constructed as was done in Fig. 5. In this case however, the vertical distance between the curve and the base line will represent the value of the e.m.f. generated by the conductor for a corresponding instant. The sine curve generated by a conductor revolving about an axis at a constant speed in a uniform magnetic field is the ideal pressure curve from an alternator and represents quite closely the e.m.f. and current waves generated by a large class of alternating-current generators.

B. O. E. Boiler-Tube Cleaner

A new boiler-tube cleaner intended for manual operation has been recently perfected and placed on the market by E. C. French & Co., 610 Monadnock Building, Chicago. It operates with a "hammer and chisel" effect; that is, it has a cutting edge that goes under the scale and forces it off. A wire brush following the cutting edges pushes out any loosened material that has failed to ride out on the knives, so that the work of cleaning is all done in one operation.

As will be apparent in the accompanying illustrations, a series of eight heat-treated tool-steel prongs are attached to the body of the cleaner. To the end of these prongs case-hardened knives are riveted, and alternate knives are ½ in. behind the others. They are self-sharpening and if necessary may be easily replaced.

A short piece of pipe passes through the body of the cleaner. One end is threaded for a pair of locknuts and a reducer to which the handle is attached. To the other end within the inclosure formed by the prongs, the driving piston is secured. The pipe is free to move through the body of the cleaner, so that a forward thrust on the handle forces the piston in contact with the prongs, pushing them out to a degree determined by the position of the locknuts.

CONTRACTED — COMBINATION BRUSH AND TUBE SCRAPER — EXPANDED

Normally the diameter of the cutting surface is contracted so that the cleaner may easily enter the tube, as indicated in the view at the left. As the brush enters the tube and more force is required on the handle, the piston is pushed forward and the knives forced outward, as indicated in the view at the right.

When the knives become worn it is easy to set the locknuts a little farther back, so that the piston may move forward an equal distance and still maintain the cutting edges at the surface of the tube. When the cleaner has reached the end of the tube its removal is easy, as a pull on the handle draws the piston away from the prongs and allows them to assume their normal position away from the tube surface.

Durability of Oil depends more on its use, or treatment, than on any inherent quality it may be supposed to possess. A rapid circulation of oil through the bearing so that it will not become overheated and vaporized is the most satisfactory and economical treatment for the oil.

Relative Value of Lubricants depends on first cost, wastage of the lubricant, difference in the power cost chargeable to them and the cost of application or attention, all of which are widely variable quantities.

The Cheapest Fuel and Most Economical Load

By Robert H. Karl*

SYNOPSIS—Results of a series of tests at the St. Louis plant of the Anheuser-Busch Brewing Association to determine the cost difference in burning screenings and nut coal and to find the most economical load for the boilers.

In the table presented herewith are the results of a series of tests conducted by the engineering department of the Anheuser-Busch Brewing Association, of St. Louis, Mo., to determine under what loads the boiler is most efficient and whether it pays to burn good but high-priced nut coal or cheap screenings. Following the table are charts or curves traced from the results obtained.

No special precautions were taken to get the boiler in extra-good condition for these tests; the one that could be used most conveniently was picked from a battery of fourteen and was given the ordinary cleaning that all

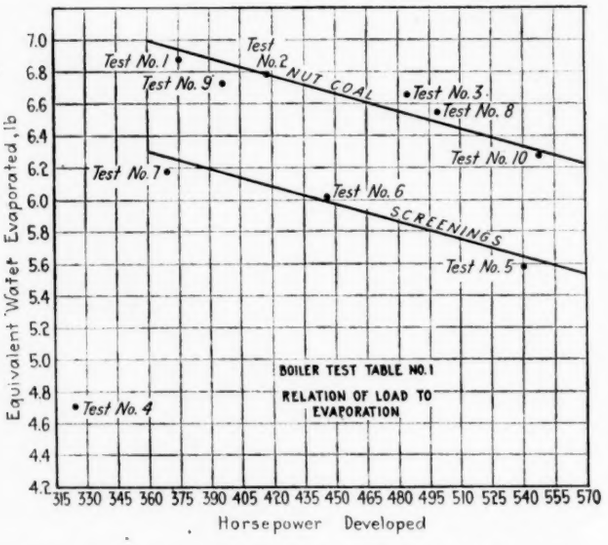


FIG. 1. EVAPORATION PER POUND OF COAL AT DIFFERENT LOADS

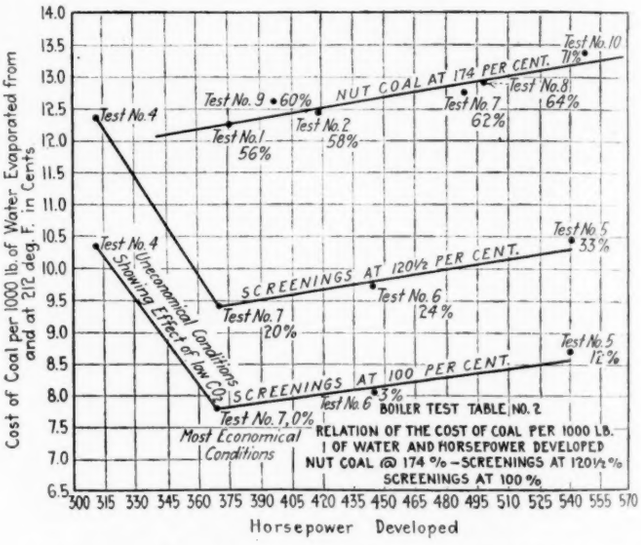

Cost of Coal per 1000 lb. of Water Evaporated from and at 212 deg. F. in Cents
Horsepower Developed
Test No. 10 71%
NUT COAL AT 17¼ PER CENT.
Test No. 9 60%
Test No. 8 64%
Test No. 7 62%
Test No. 4
Test No. 2 58%
Test No. 1 56%
Test No. 5 33%
SCREENINGS AT 120½ PER CENT.
Test No. 6 24%
Test No. 7 20%
Test No. 4
Showing Effect of Low CO2
Test No. 5 12%
SCREENINGS AT 100 PER CENT.
Test No. 6 3%
Test No. 7, 0%
Most Economical Conditions
BOILER TEST TABLE NO. 2
RELATION OF THE COST OF COAL PER 1000 LB. OF WATER AND HORSEPOWER DEVELOPED
NUT COAL @ 17¼ % — SCREENINGS AT 120½ %
SCREENINGS AT 100 %

FIG. 2. SHOWING THE RELATIVE COSTS OF EVAPORATION WITH SCREENINGS AND NUT COAL

of our boilers get. Gages were tested, thermometers inserted in the feed-water line close to the boiler, a calorimeter was placed in the steam outlet, the feed water piping was rearranged especially for the tests, and the boiler was operated ten days before the tests were started, the purpose being to get average daily operating conditions. As this series of tests extended over a period of three weeks and as our feed water makes considerable scale, the boiler kept getting dirtier as the tests went on, which was the condition wanted.

This plant is equipped with all instruments necessary for the economical combustion of fuel, such as recording feed-water thermometers, recording feed-water meters, differential draft gages on each boiler, together with recording draft gages, recording electric pyrometers for taking flue-gas temperatures, and CO_2 recording machines. During the tests the recording CO_2 machines were checked with an Orsat. Tanks specially built for weighing feed water were installed, and a separate feed-water pump was used.

Increasing the load on a boiler will decrease its efficiency because a larger amount of coal will be burned, evolving a larger volume of gas, which, to be discharged, will necessarily have to pass through the boiler at a greater velocity and will be unable to have proportionately the same amount of heat absorbed from it. The greater the load on the boiler, the greater will be the flue-gas temperature and consequently the lower the efficiency, as a larger amount of heat in the gases will escape up the chimney.

The increase of the flue-gas temperature is partly due to increased volume and greater velocity, as already explained, and partly to larger evolution of volatile matter, making it impossible to burn it all in the furnace, transferring the point of complete combustion further into the chamber or among the tubes. The nearer this point is to the uptake, the higher will be the final temperature.

Usually, the most economical way of working a boiler is at its rated capacity, for increasing the rate of steaming increases the final temperature, and by decreasing the rate, if the temperature will not go up directly, it at least will not go down proportionately.

Probably the best explanation for this is that with low rates of combustion the motion of the gas is comparatively slow and the friction between the gas particles is greatly diminished. Next to the heating surface is a thin film of dead gas, and the slower the velocity the thicker will be this film; and as gas is a poor conductor of heat and as heat is not transferred to the boiler surface by conduction but by convection, this film will keep the heat of the newly made hot gas away from the metal.

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BOILER TESTS MADE AT THE ANHEUSER-BUSCH BREWING ASSOCIATION BY THE ENGINEERING DEPARTMENT

Heine water tube boiler No. 1; rated Horsepower, 334; Green chain-grate stoker. Heating surface, 334 sq.ft.; effective grate surface, 72 sq.ft.; ratio of heating to grate surface, 46.3.

	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8	No. 9	No. 10
Number of test										
Date	Feb. 24, 1915	Feb. 25, 1915	Feb. 16, 1915	Mar. 3, 1915	Mar. 5, 1915	Mar. 6, 1915	Mar. 9, 1915	Mar. 10, 1915	Mar. 11, 1915	Mar. 12, 1915
Duration of test	Eight Hr., 55 Min.	Seven Hr.	Eight Hr.	Eight Hr.	Eight Hr.	Four Hr.	Eight Hr.	Eight Hr.	Eight Hr.	Eight Hr.
Coal:										
Kind of coal	Nut coal	Nut coal	Nut coal	Screenings	Screenings	Screenings	Screenings	Nut coal	Nut coal	Nut coal
Condition of coal	Good, 85% fine	Very wet, rain	25% fine, very wet	25% fine	50% fine	Good, 60% fine	Good, 40% fine	75% fine	60% fine	60% fine
Coal consumed, actual, lb	15,044	14,870	20,302	18,265	26,627	10,266	16,444	21,091	16,255	28,980
Coal per hr., (dry), lb	219	26.26	29.36	27.09	40.2	30.7	24.9	32.3	25.4	37.3
Coal per sq.ft. grate sur. per hr. (dry), lb	0.485	0.566	0.632	0.58	0.866	0.66	0.536	0.696	0.54	0.805
Coal per sq.ft. heating sur. per hr. (dry) lb										
Analysis of Coal:										
Fixed carbon, per cent	36.72	35.71	36.33	34.79	34.80	34.71	34.09	34.16	36.50	34.93
Volatile matter, per cent	34.09	35.74	34.92	32.22	34.07	32.58	34.73	33.47	36.51	36.52
Moisture, per cent	12.13	12.06	14.09	14.32	13.01	13.74	12.76	11.33	9.79	10.29
Ash	17.06	15.59	14.66	18.47	18.12	18.97	18.42	21.04	17.20	18.25
B.t.u. actual	10,034	10,107	10,322	9,402	9,768	9,251	9,779	9,744	10,479	10,228
B.t.u. dry	11,199	11,612	12,015	10,999	11,229	10,725	11,210	10,990	11,616	11,338
Ash:										
Weight of ash, lb	2,555	2,578	3,996	3,636	4,988	1,971	3,025	4,355	3,479	4,365
Fixed carbon (combustible), per cent	25.66	23.95	20.93	21.78	18.32	19.78	16.56	15.23	32.12	20.9
Ash in ash, per cent	74.13	76.05	79.07	78.22	81.78	80.22	83.44	84.77	68.88	79.1
Water:										
Water per sq.ft. heating sur. per hr., lb	3.76	4.326	5.06	3.21	5.23	4.63	3.81	5.16	4.09	5.65
Steam:										
Quality of steam per cent. (dry)	99.4	99	98.9	99.04	98.92	98.94	98.07	99.08	98.8	98.86
Total weight of moisture in steam	937	958	1,421	771	1,540	618	995	1,289	1,051	1,616
Pressures and Draft:										
Steam pressure, lb	110	110	109	107	108	108	109	108	108	111
Draft in uptake, inches of water	0.17	0.30	0.53	0.56	0.85	0.67	0.28	0.69	0.19	0.91
Draft over the fire, inches of water	0.08	0.14	0.23	0.26	0.33	0.26	0.12	0.26	0.09	0.34
Fuel thickness in inches	5	4½–5	5½	4½–5	6	5	5½	5½	4½–5	6
Temperature:										
Feed water, deg. F	182.6	187	196	186	196	184	189	196	188	190
Flue gas, deg. F	585.8	650	667	605	683	608	677	677	625	705
Outside air, deg. F	50	50	55	37	58	58	40	40	45	58
Horse power:										
Developed horsepower	375.7	418	480	311	540	445	369	499	396	546.7
Percentage of rated horsepower	112	125	146	93	162	133	110	149	119	164
Cost of Coal:										
Cost of coal per ton in car, dol	1.695	1.695	1.695	.975	1.175	.975	1.175	.925	1.695	1.695
Cost of coal per 1,000 lbs. of water actual, dol	.1303	.1318	.1346	.1108	.1330	.09203	.1108	.0866	.104	.0837	.1066	.1335	.1433
Cost of coal per 1,000 lb. of water equivalent, dol	.1226	.1245	.1273	.1037	.1243	.0870	.1048	.0869	.0975	.07866	.09479	.1261	.1346
Cost of coal per 1,000 lb. of water equivalent, dol	.00433	.00439	.00439	.00357	.004253	.00300	.003616	.003323	.00392	.00271	.003272	.00445	.004647
Cost per horsepower										
Results:										
CO_2, per cent	12.3	11	11.5	8	10.6	9.9	10.6	10.4	11.1	10.1
Flue-gas temperature, deg. F	585.8	650	667	605	683	608	614	677	625	705
Horsepower developed	375.7	418	480	311	540	445	369	499	396	546.7
Water evaporated per lb. of coal actual, lb	6.42	6.37	6.20	4.41	5.28	5.62	5.82	6.20	6.34	5.91
Water evaporated per lb. of coal equivalent, lb	6.88	6.79	6.65	4.70	5.58	6.01	6.19	6.55	6.73	6.28
Efficiency commercial (dry coal), per cent	67.82	66.6	64.2	50.4	57.1	64.9	63.28	66.6	63.7	61.0
Efficiency available (actual coal), per cent	66.6	65.3	62.5	48.5	55.6	63.1	61.5	65.2	62.66	59.7

Average B.t.u. nut coal actual 10,169. Average B.t.u. screenings actual 9,550. Average B.t.u. nut coal dry 11,461. Average B.t.u. screenings dry 11,040. No. 8 nut coal compares with screenings.

Chart No. 1 shows how the evaporation per pound of coal decreases as the rate of working the boiler increases. It also shows the value of screenings as compared with nut coal. The cost per ton of screenings was 44 per cent. less than per ton of nut coal, and the value as a steam producer was only slightly over 9 per cent. less.

Chart No. 3 shows the increase of the flue-gas temperature with the increase of horsepower developed and proves the foregoing discussion. It also shows the theoretical loss due to the increase of flue-gas temperature. Supposing the percentage of CO_2 to be 10 per cent. for all tests, it proves that even with high CO_2 for both tests, the efficiency of the boiler under No. 10 test conditions can be increased 6.25 per cent. by changing it to conditions of No. 1 test.

The rate of working a boiler should be regulated only by the amount of draft carried. To maintain economical conditions a draft gage on the front of the boiler is necessary.

As to the amount of draft that should be carried, it is impossible to lay down any hard and fast rules, because this depends upon the draft obtainable, the kind of fuel, the fuel-bed thickness and the ratio of heating to grate surface. In many plants the grate surface under the boilers is much too large, making it impossible to reduce the draft sufficiently to bring the working of the boiler down to rating without causing smoke and incomplete combustion. Generally speaking, carry the draft as low as possible and still keep up the steam pressure, but not low enough to make smoke.

To have low flue-gas temperature is important, but still more important is the CO_2. Of all the tests made, No. 4 showed the least economy. During this test only 8 per cent. CO_2 was obtained on the average. Now, 8 per cent. CO_2 indicates the use of about 30 lb. of air per pound of coal, while only 20 lb. should have been used. The theoretically computed loss that could have been prevented was 11 per cent. of the coal fired, while the actual loss as shown by comparing the figures with other tests amounts to nearly 20 per cent. Very small plants should have portable hand machines, and in large ones it would pay to have a recorder for each boiler.

In many plants the firemen are overcareful to not lose too much unburned carbon in the ashpit. This occurs es-

pecially in plants with chain-grate stokers, with the result that to prevent this loss the fires are carried short. It is of course not necessary to run an unduly large percentage of carbon over into the ashpit, but the grate should be covered at all times. The slightly increased loss in the ashpit will be more than compensated for by the saving effected by reducing the unnecessary excess air.

The higher the draft, the more difficult it will be to keep a high CO_2. With a high draft every opening in the furnace will be admitting a correspondingly large amount of

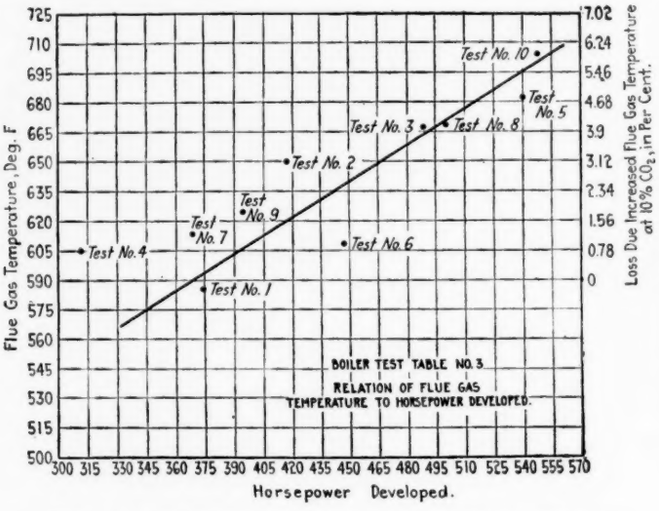

Flue Gas Temperature, Deg. F
Loss Due Increased Flue Gas Temperature at 10% CO_2, in Per Cent.
Test No. 10
Test No. 5
Test No. 3
Test No. 8
Test No. 2
Test No. 9
Test No. 7
Test No. 4
Test No. 6
Test No. 1
BOILER TEST TABLE NO. 3
RELATION OF FLUE GAS TEMPERATURE TO HORSEPOWER DEVELOPED
Horsepower Developed.

FIG. 3. RELATION OF FLUE-GAS TEMPERATURE TO OUTPUT

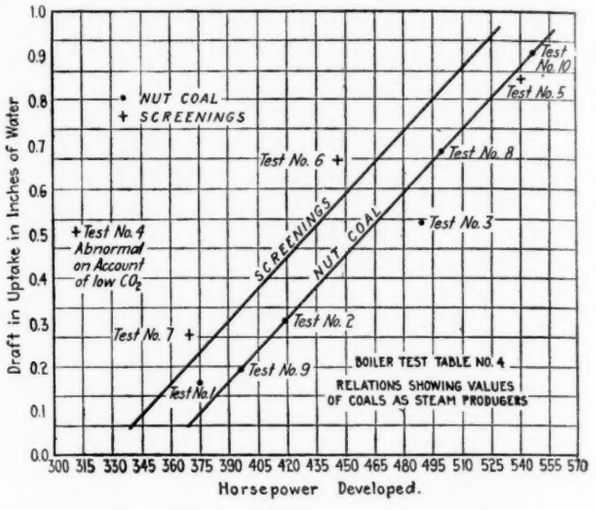


FIG. 4. SCREENINGS AND NUT COAL COMPARED AS STEAM PRODUCERS

air or leakage. By stopping leaks in the settings, increasing fuel thickness and exercising more care, as high a CO_2 percentage can be obtained with 1 in. of draft in the uptake as with 0.3 in.

Chart No. 2 shows the relative prices of the coal indicated in percentages, taking the cheaper screenings as 100-per cent. cost coal, the higher-priced screenings as 120½ and the nut coal as 174 per cent. These percentages indicate their relative cost in dollars and cents. The 100 and 120½-per cent. cost coal are the same grade of coal, the two different values being given because of variable market fluctuation of prices.

The chart shows that the most economical test made with nut coal was fully 24 per cent. higher in cost per 1,000 lb. of water evaporated than the most uneconomical test with screenings at a present market value of 120½ per cent. per ton. It also shows how the cost per 1,000 lb. of water increases as the horsepower increases, with the exception of No. 4, in which the cost greatly increases with decrease of horsepower, on account of uneconomical conditions as indicated by low CO_2.

Chart No. 4 shows the intensity of the draft required for the production of steam with the two kinds of coal. Screenings, being finer, required about 0.12 in. more draft than the coarser nut coal for the production of the same amount of steam.

In most cases the cheaper coal is the more economical to burn, but nevertheless a man, before buying, should take a number of factors into consideration: Whether the plant has a sufficient boiler capacity to burn the cheaper fuel and still make the required amount of steam (a greater weight of cheap coal will have to be burned to generate the same amount of steam); whether there will be too great an increase in labor for handling the cheaper coal and ashes; whether wear and tear of coal- and ash-handling machinery, extra freight charges and attention to the furnace will offset the saving.

In short, to direct and operate a boiler plant economically, we must burn the coal right—must have the lowest possible flue-gas temperature, about 13 per cent. CO_2, no smoke, and no carbon monoxide in the flue gases. The loss of unburned carbon in the ashpit should be the lowest possible without lowering the CO_2. To obtain all this, the use of gas-analysis instruments, CO_2 recorders, flue-gas thermometers, or pyrometers and draft gages is necessary, and such instruments should not be considered a luxury, but an absolute necessity, and any reasonable expenditure for them will amply repay itself.

Cooling Ponds for Condensing Engines under average conditions have lowered first cost, require less power for operation and practically no maintenance as compared with cooling towers, and they are being used to a constantly increasing extent in connection with cotton mills, except in cases when sufficient pond area is not available, either on the ground or on the roofs of buildings adjacent to the condensing installation. Some mills of comparatively small capacity are now operated noncondensing because of the lack of a sufficient supply of cool water. Many of these mills could install an earthen pond or a concrete basin at small expense and thus make a net saving in power cost of 15 to 20 per cent. In addition to a number of other economic advantages, the first cost of spray systems, including ponds, generally averages one-half the cost of cooling towers of similar capacity and of good design and construction.—"Journal of the Franklin Institute."

Analysis of the Products of Combustion will indicate the method of stoking that produces the best results. As is well known, the carbon in coal is never completely burned to carbonic-acid gas but is often changed only into carbon monoxide. Even the carbon dioxide that is formed often prevents complete combustion if it comes in contact with overheated coal. It then forms carbon monoxide as it passes through the flues and chimney. If a pound of dry coal burned entirely to carbonic-acid gas yields 14,600 B.t.u. that weight of the same coal burned, only to carbon monoxide will give but 4,450 units of heat. If an engineer could, by using a carbon-dioxide recorder, increase the percentage of carbon dioxide in the exit gases from 5 to 14 per cent., it would effect a saving in coal of 21½ per cent. If, in addition to this, he succeeded in reducing the temperature at the stack 100 degrees, the saving would amount to 24 per cent.—"The Boilermaker."

The First Shipment of Illinois Coal was made in 1810, in which year a flatboat load was sent to New Orleans from Brownsville, Jackson County.

Editorials

Practicing Voluntary Economy

An engineer can invariably get plenty of pleasure out of his vocation, no matter where he is located. An engineer in a small town out west has run the village pumping plant and electric-light plant for a score of years.

The village employing this man does not know what a good man he is compared with other engineers that might have been employed in his place. There is no central station near-by to compete with his rates and force him to economize, but he economizes anyway. The people of the village know nothing about power, rates or competition, nevertheless he economizes. His equipment is about the crudest—noncondensing, simple engine, old-type boilers, etc. In spite of this also he economizes.

Most other engineers would pay no attention to the cost of a horsepower-hour in such a plant as his. But this engineer knows. He knows how much water each pound of coal evaporates; the efficiency of his boilers, of which he has only two (and he knows which of the two is the more efficient); the indicated, friction, and brake horsepowers of his engine; the transmission efficiency of his belt; the efficiency of his generation; his line losses, etc. He has made a careful study of every detail of his plant, and every detail shows it. One can see with a single glance that there is a master hand about the plant. Everything is shipshape, clean, polished and "efficient looking." He is proud of his plant, and he has reason to be.

His salary is not large, but he does not seem to care: it is sufficient for comfortable living in that village, and he is so well rooted there he would stay, probably, even if his pay were reduced.

The principal point is, he gets pleasure out of running his plant as efficiently as possible, even though not compelled to do so.

⁂

Boiler Inspection in Colorado

Whether 'tis better to maintain a boiler-inspection department when the treasury is low or make the factory inspectors do such duty—that is the question that Governor Carlson of Colorado has decided to his own satisfaction and to the displeasure of many citizens of that state of turbulent memories.

It was believed by the General Assembly that an emergency existed, and an appropriation totaling $19,600 was proposed for 1915 and 1916, but the governor saw fit to cut it nearly seventy-four per cent., or to a beggarly $5,150. And, mind you, the items consisted only of salaries and traveling expenses.

The governor pleaded economy. It seemed a waste of time and money to have two forces of inspectors visiting the same usually widely scattered plants. Why could not the factory inspectors inspect boilers as well as fire escapes and what-nots? But let us take up that question later. Colorado charges five dollars for each internal and external inspection, and unlike some states and cities, this charge applies for any number of boilers inspected at one time. *Power* does not know how many boilers subject to inspection there are in Colorado exclusive of those in Denver, which inspects its own. But there were 1,496 inspected in that city in a recent year. Denver may have about one-third of the boilers in the state; it is not likely that it has much more. And at five dollars per boiler 3,940 inspections would be enough to put back into the treasury an amount equal to the proposed appropriation. And here is the point: The state would be assured of competent inspection and the upholding of the splendid record of the boiler-inspection department, which, the *Denver Post* says, has not had an explosion in two years. Ordinarily one does not expect a purely safety measure to be entirely self-supporting.

State finances and the fact that the boiler plants of Colorado are far apart over an extensive area not overcrowded with railroads and other means of transportation may make it expedient to have one force of inspectors to care for both factories and boilers; but if this is so let the inspector first be a competent boiler inspector. He is fitted by rigid training with the qualities demanded of a factory inspector, and the knowledge of that occupation he can much more easily and quickly acquire than the factory inspector can a working knowledge of boiler inspection.

⁂

Engineering a Municipal Plant

When one has read the article on the Regina Municipal Power Plant, it would be well to re-read Professor Hirschfeld's description of the Connor's Creek Power Plant at Detroit, in *Power* of September 14, 1915. Some interesting comments can be made on these two plants. The Detroit station may be considered as the "last word" in American power-plant design, while Regina is probably the best example of a high-grade Canadian municipal central station. Both designs were largely made by the staffs of the two corporations, but in Detroit the engineers worked from their experience with the large Delray station, which was already in service.

At Regina, the superintendent, as a basis for his designs, had only his experiences with a small old plant not built on modern lines and with heterogeneous equipment. The results of his efforts become of greater interest to the average engineer when he learns that this splendid plant was designed and built by a fellow engineer who, without technical training, worked up from the ranks to his present position of responsibility. Such results could only be obtained by close and careful study of modern development in current literature and by extended trips to the large stations in the East.

Comparing the equipment details of the two plants, one finds some points of similarity. Both use comparatively high-pressure steam in their boilers. They also employ underfeed stokers with large combustion chambers. Coal and ash are handled mechanically. The electric drive is used quite generally for all auxiliaries. Both use

forced draft under the grates; the Regina employs induced draft also in place of a chimney.

Steam turbines with surface condensers are common to both plants, though of course of different size. Regina employs vacuum augmenters to obtain high vacuum. It seems rather remarkable that both stations should be built simultaneously and independently and yet each contain innovations in the use of the discharge air from the turbo-generators as a supply for the stoker fans.

At Detroit the feed-water temperature is regulated by a special barometric condenser. In the Regina plant certain auxiliaries can be either steam- or motor-driven, depending on the amount of steam required for feed-water heating.

Following English practice, Weir feed pumps are used in place of turbine-driven centrifugals at Regina. With coal costing $5.50 a ton, every device was necessary to obtain the highest possible amount of heat from the fuel, and hence economizers in a novel setting were installed.

To some it may seem impertinent to thus compare the two stations. The real object of these paragraphs is to call the attention of those in charge of electric plants to the commendable results obtained by an engineer in the service of the public, in the construction of a plant embodying uptodate ideas and situated in a section of the country where one would least expect such development.

Determining the Value of Coal

"Are we using the right coal?" is a question frequently coming up in the power plant. All too many, from a lack of knowing how or from the want of facilities for testing their coal, are content to let well enough alone. They have been getting along nicely with the kind and size they have been using and feel that the safest course is to continue to specify the same. Without doubt this is wiser than indiscriminate experimenting, especially when the latter is provoked merely by a desire to find a fuel that is cheaper in purchase price. The real value of a coal is measured in the cost of the steam produced with the given boilers and furnaces. One cannot rely on the experiences of other plants with certain coals as a sure or even probable indication that a like success will follow in his adoption of the same fuel. Even if his boilers are similar his furnaces may differ in proportions, in grates and in draft afforded, and either one or all of these conditions may act to render the promising coal unsuitable.

It must be remembered, then, that boiler furnaces are designed with a view to the kind of fuel to be burned, but size alone is not the controlling factor because a certain form of grate bar is used. If the percentage of ash is not that adapted to the proportion of grate area provided, the coal is not the best even though of proper size.

All of this may seem an argument for the stand-pat policy of adhering to one kind of coal that has been satisfactory in the past, but the point is that, while little appreciated by most purchasers, there may be a considerable variation in the ash content of coal from the same section of the country and even the same mine, and to such an extent as to markedly affect its value. While it is not advisable on general principles to go far afield in the trial of a different grade of coal or one from another locality, it is well to keep careful watch on the supposedly good coal that one is buying. If the ash content is found excessive, penalties should be imposed on the coal supplyman.

Some three years ago there was printed in these columns the report of the Prime Movers Committee of the National Electric Light Association, which contained a simple hint on determining the value of a coal—a hint well deserving of repetition. It was this, that one and one-half times the percentage of ash and not the percentage itself alone should be deducted from the one hundred per cent. value of the coal. Particularly where a calorimeter test is not possible or convenient (and at times even such a test may be misleading), the foregoing is a simple and easily remembered rule that should prove helpful to those concerned with the purchasing of coal.

Although variation in ash alone is not a sure measure of the value of a coal, still, under the limited conditions that obtain in any individual case it is quite likely to be reliable, and at least is an attempt to do something toward securing a better economy than is apt to result where no endeavors are made to check up the worth of the coal bought. It is so simple a precaution that scarcely any plant has excuse for neglecting it.

Central Station Acquires a Block Plant

Word comes that the Mormon Church has sold out a one thousand horsepower block plant, which supplied power, light and heat to a number of its buildings, to the Utah Power and Light Company. The building containing the plant was the Hotel Utah. A member of the committee handling the transaction and acting for the hotel company advises that the transfer was made in the interest of service. Quoting from the Salt Lake *Tribune*, "it is held that the growth of the power industry has been so swift in recent years that only a company devoted exclusively to the furnishing of power can keep pace with improvements." (Central-station solicitors please copy.)

As it is our business to keep informed upon and record the advance in the power field, it comes as something of a shock to us that the advance has been so swift that a plant installed in 1911 is already out of date. We did not realize it.

The account goes on to say that a deal for purchasing power was made for a term of years "satisfactory to both contracting parties." From its reputation for shrewdness in money affairs, we are inclined to think that the Mormon Church lost nothing by the change.

Now, as the plant was relatively new and the block plant as a type is the most economical, what is the inference? None other than that the central station—and this is the significant thing—is so afraid of block-plant competition that it will go to any lengths to remove it.

In dealing with alternating currents the sine curve is as common as the steam tables are in steam engineering; but how many who have not received technical training really know what the sine curve represents and how it is evolved? In the series of articles which begins in this issue, Mr. Annett has endeavored to present some of the fundamentals commonly met in alternating-current work, in such a way that it will be easy to get a clear conception of their meaning and applications.

Correspondence

Cannot Increase the Vacuum

I recently visited an engine room at Lynchburg, Va., where I found a Corliss engine operating with a jet condenser. The vacuum gage on the system indicated but 23 in. of vacuum. The engineer stated that upon taking charge of the plant a few months previously, he had found the condenser down and out and the engine operating noncondensing. He had overhauled the condenser and put it in operation and had been running condensing for several weeks, but after the load came on he could not get more than the 23 in. of vacuum, although when first starting up with the engine running light, 26 in. could be obtained.

An increased amount of injection water was suggested, but this had been tried with no additional gain in the vacuum. Leaky connections between the condenser and the engine were also mentioned as a possible cause, but the engineer said that he had gone over all joints and that they were tight, as was also the valve-stem packing on the engine-exhaust valves; the back-pressure valve was likewise tight.

It was then suggested that perhaps the vacuum gage was inaccurate and that the condenser was maintaining a higher vacuum than indicated. The engineer did not know as to the correctness of the gage, but would have it tested to make sure.

Perhaps some reader can suggest a reason for the low vacuum during the period the engine carries the heaviest load, which is about three-quarters of the engine capacity.

New York City. R. O. Warren.

Starting Up a Heating System

A heating system that has been laid up for the summer is seldom in condition to put into service in the fall without a thorough going over.

If thermostatic traps are used they will frequently be found stuck, either shut or open—usually the latter. Oily deposits, rust, etc., when hardened, will render them inactive. If stuck open they will form a "short-circuit" from the supply to the return line and render the rest of the circuit useless, as well as permitting steam to be drawn through the return line to the vacuum pump. In order to produce a vacuum in such a case an over-amount of spray water must be introduced, which will cause the heater (if an open one is used) to overflow to a wasteful extent. Daily overflowing of open heaters is good practice, however, where cylinder oil comes back in the return pipes, as the oil may thus be flushed over to the sewer before it has time to emulsify with sinkable matter held in solution in the feed water. Much of the oil may be kept out of the boilers by this means.

In order to have a dry, quiet, reliable system where radiators are used, absolute tightness is necessary. One should be able to get the same vacuum at the farthest extremity of the return line as at the pump. If every unit is equipped with a perfectly operating non-vapor-passing trap and a sufficient vacuum is maintained to remove all condensate promptly, it will require a very slight steam pressure to creep around.

The danger of water-hammer while warming up a long circuit is greatly lessened by working up a good vacuum before hand. Plenty of time is to be a watchword in turning in the steam to a cold line or system. Otherwise the return pipe lines will be overtaxed by the rapid condensation and water-hammer will be set up, probably resulting in disaster.

Where exhaust steam is to be used, I prefer to "warm up" with live steam in advance, as it is easier to feel one's way. Edward T. Binns.

Philadelphia, Penn.

Experiences of a Refrigerating Troubleman

In my article on this subject in the Sept. 7 issue of *Power* it was not my intention to convey the idea that cleaning double-pipe condensers with sand and water is bad or is incorrect engineering. Cleaning double-pipe condensers with sand and water under pressure is in the same category with many other engineering kinks for doing things easily and cheaply—good if properly done and bad if not. In the case I mentioned, the condenser was not properly washed free of sand, consequently some of it found its way into the compressor cylinder, causing scoring and excessive wear, which necessitated reboring of the cylinders.

Mr. Blanchard's comment on this article (*Power*, Oct. 12) indicates that he does not believe that sand could pass through the evaporator coils and into the compressor. If he will notice the contents of the scale trap on the suction line this winter when resuming operations after overhauling and cleaning the evaporator coils, he will probably find large pieces of scale that have passed from out these coils (under normal working conditions) into the suction line and lodged in the trap. Upon examination these pieces of scale will be found to be much heavier and sometimes larger than a dozen grains of sand. In view of the foregoing it is an easy matter to see how grains of sand travel through the system and the suction line or scale trap and finally enter the compressor cylinders.

On page 521 of the Oct. 12 issue, Mr. Thurston has calculated the volume of an 11x24-in. ammonia compressor and the speed such a volume of liquid has to attain to be discharged through a 1-in. extra-heavy pipe in a certain time. The figures are correct, but I have never seen an ammonia compressor that was 100 per cent. efficient as a liquid pump, as Mr. Thurston has assumed.

It is quite evident that he is basing his statement on experience with a 40-ton plant. There is a difference between a 40-ton plant and a 5-ton plant. The plant that I described had a belt-driven 5-ton vertical twin-cylinder single-acting ammonia compressor with false heads acting

as discharge valves. It was operated at 35 to 40 r.p.m., this low speed being necessary as the compressor is too large for the work it has to do.

The condenser, liquid receiver and double-pipe brine cooler are nearly all on the same level and hung on brackets on the wall around the compressor. Of course I did not suppose that anyone would get the impression that this machine was 100 per cent. efficient as a liquid pump and was pumping a cylinderful of liquid at each stroke. This compressor has false heads as discharge valves; this feature and the slow speed would allow the compressor to pump some liquid without any appreciable pounding. The compressor did not, nor would it, pump a cylinderful of liquid or even a half cylinderful at every stroke, but this type will handle much liquid without danger and with little noise. I also know from experience that this compressor will pump so much liquid, when the evaporator coils are full, that the pulsations and the strokes can be observed in the liquid-receiver glass as the liquid rises and falls with every stroke of the compressor. C. E. Anderson.

Chicago, Ill.

In the Oct. 12 issue of *Power* Thomas G. Thurston, in commenting on "Experiences of a Refrigerating Troubleman," by C. E. Anderson, in the issue of Sept. 7, goes into the subject of an ammonia compressor acting as a liquid pump. I do not believe that Mr. Anderson meant that the compressor was handling liquid, but that the vapor was so heavily saturated that it contained more than the usual amount of liquid.

An ammonia compressor will handle vapor that contains enough liquid to cause pounding and racing. I have never known of a machine being wrecked by the presence of liquid. Slugs of liquid, such as are often drawn in from the evaporating coils, may cause a suction valve to break, and this will result in an accident; but the chance of a really serious accident caused by liquid in the compressor is slight.

If a machine is operating on a low back-pressure, the presence of liquid will be discovered in time to check it at the suction stop valve or to stop the machine. If the back-pressure is high, the liquid entering the compressor will cause the machine to slow down or stop. Ammonia compressors are now equipped with safety valves, and these will give ample warning when too much liquid is getting into the cylinder. A belt-driven machine will most likely do more damage to the belt than to anything else if an overdose of liquid is drawn in. The compressor will stop and the motor will keep going (if the belt is at all loose) and the belt will burn.

In large direct-expansion systems the presence of liquid with the vapor is a constant source of annoyance. At times the machines have to be slowed down and the suction valves partly closed while the machines are "pounding through" an unusually bad dose. But accidents directly caused by the liquid are rare.

Referring to Mr. Anderson's article of Sept. 7, it looks hardly possible to operate a machine under the conditions stated. With 60 lb. back-pressure I do not believe an ammonia machine of any ordinary type will run. The load will be too great for the engine or motor to pull. If it is steam-driven, the engine will run slow and take steam the entire stroke. If motor driven, the fuse will blow, the circuit-breaker jump out, or the belt slip and burn. Such are the things I have had happen whenever the back-pressure got high, and I am sure they happened long before 60 lb. was reached.

Consider the difference in the weight of ammonia pumped when the back-pressure is the usual 20 lb. and when it is 60 lb. Look at the ammonia tables and the following will be found: Saturated vapor at 20 lb. pressure weighs 0.1247 lb. per cu.ft.; at 60 lb., it weighs 0.2575 lb. The weight of ammonia pumped at 60 lb. pressure is more than twice that at 20 lb. The engine or motor will not pull the machine at such an overload.

If the compressor has been shut down for several hours, and if it happened that the ammonia had been allowed to get into the evaporating coils and give a pressure of 60 lb., the suction stop valve could not be opened more than about one turn until the machine had again been in operation long enough to reduce the pressure. The machine could not be started with the suction valve open, and it would stop again if the valve were opened wide.

The high back-pressure found on the low-pressure gage when a machine has been shut down over night shows the pressure caused by the light superheated gas in the coils. Two or three revolutions of the machine gets rid of this. There is much difference between saturated vapor and superheated gas. A. G. Solomon.

Chicago, Ill.

Central-Station Rate Making

Most discussions of rates for light and power fail to consider whether the rates suggested will get and hold the business; especially is this true of the power load. If the power business cannot be supplied at the rates evolved, it is evident that there will be no power business. For in these days of keen competition, if a prospective power consumer cannot be convinced that he will have to pay out little more for his central-station service than for service from his own plant, an isolated plant will be installed. It was therefore with a great deal of interest that the writer as a power salesman read the article on "Central-Station Rate Making" by Paul J. Kiefer in the Aug. 24 issue.

While it is impossible to say whether or not the 100-, 200- and 500-kw. consumers could be supplied at the rates deduced in the particular locality which Mr. Kiefer had in mind, it is certain that the central station would not be able to hold this business in many localities where the central-station costs would be about as shown. Assume that the central station is unable to get the business at these rates and that these consumers have installed isolated plants, and determine the costs to the small consumer.

The size of the plant would be about the same as that given as the sum of the lighting maxima exceeds 3,000 kw., and at certain periods of the year these maxima occur at the same time. The cost would probably be reduced $50,000 in line extensions, transformers, etc.; the other costs would remain as before. Meter reading, building, etc., would also remain practically constant, as only 19 out of 3,450 consumers would be done away with.

The expense of fuel, water, lubricants, waste, station labor, etc., can be divided into two items, one representing fuel and water and the other covering the remaining items, these being about $55,000 for the former and

$30,000 for the latter. Under the load conditions assumed, the fuel cost amounts to 0.659c. per kw.-hr. On account of the decrease in load factor, due to the power business dropping off, this would be increased about 25 per cent., or to 0.824c. per kw.-hr., or for the 4,338,000 kw.-hr. used by the lighting and small power users, $35,747.12. The item of station labor, etc., would remain practically the same, possibly dropping off $1,000, or to $29,000.

Following is a résumé of the foregoing:

Capacity of plant	3,000 kw.
Cost of station, offices, lines, etc.	$750,000.00
Yearly expenses:	
Station and system upkeep, administration, superintendence, general, legal and advertising expenses	$75,000.00
Interest at 5 per cent.	37,500.00
Depreciation, taxes, insurance at 7 per cent.	52,500.00
Meter reading, billing, etc.	10,000.00
Additional profit at 3 per cent.	22,500.00
Fuel, water, lubricants, labor, etc.	64,747.12
	$262,247.12

Computing the three items of the total charge as before,

$$\text{Consumer's charge } \frac{10{,}000}{3{,}431 \times 12} = 24.3\text{c. per month.}$$

$$\text{Demand charge } \frac{165{,}000}{4{,}605 \times 12} = \$2.95 \text{ per kw. demand per month.}$$

$$\text{Energy charge } \frac{87{,}247.12}{4{,}338{,}000} = 2.01\text{c. per kw.-hr.}$$

CHARGES TO CONSUMERS

	Individual Monthly Charge				Rate per Kw.-Hr.
	Consumer	Demand	Energy	Total	
Residence:					
1-kw. connections	$0.243	$1.915	$0.543	$2.701	$0.10
2-kw. connections	.243	2.950	1.04	4.233	.0814
Commercial:					
6-kw. connections	.243	15.92	6.04	22.203	.074
15-kw. connections	.243	39.85	20.10	60.193	.0602
30-kw. connections	.243	79.70	50.30	130.243	.0521
Street lighting:					
640-0.625-kw. lamps	...	1.84	3.77	5.61	.0295
Commercial motors:					
20-kw. connections	.243	35.40	20.10	55.743	.0557
40-kw. connections	.243	70.80	40.20	111.343	.0557

It will be noticed that in order to pay expenses and give the same rate of return on the investment, it will be necessary to charge the small consumer a much higher rate. On the other hand, if it is decided that in order to get the large power business it is necessary to arbitrarily cut the demand charge to, say, $1.10 per kw. of maximum demand per month, what would be the result? According to the theory advanced there would be no justification in so doing, but if it would result in a lower net rate to the small consumer, would it not be advisable to do this?

The total monthly demand charge paid by the 19 large consumers in the article mentioned was $4,217.70, or $50,612.40 per year. Therefore the demand charge to be paid by the large power consumers will be 1,936 × $1.10 × 12 = $25,195.20, leaving $145,804.80 as the demand charge to be paid by the small consumer, or $2.63 per kilowatt of maximum demand. The total charges will then be

	Individual Monthly Charge				Rate per Kw.-Hr. Cents
	Demand	Consumer	Energy	Total	
Residence:					
1-kw. connections	$1.71	$0.2415	$0.3521	$2.3036	8.54
2-kw. connections	2.63	.2415	.6781	3.5496	6.57
Commercial:					
6-kw. connections	14.21	.2415	3.9124	18.3639	6.12
15-kw. connections	35.505	.2415	13.0414	48.7879	4.88
30-kw. connections	71.01	.2415	32.6034	103.8549	4.15
Street lighting:					
640-0.625-kw. lamps	1.64	....	2.4452	4.0852	2.16
Commercial motors:					
20-kw. connections	31.56	.2415	13.0414	44.8429	4.48
40-kw. connections	63.12	.2415	26.0828	89.4423	4.47
Wholesale motors:					
100-kw. connections	60.50	.2415	104.3311	165.0726	2.06
200-kw. connections	116.60	.2415	234.7450	351.5865	1.95
500-kw. connections	275.00	.2415	639.0281	914.2696	1.86

From the foregoing it is to be seen that the charges are less to the small consumers than if the cost to the power consumers had not been lowered.

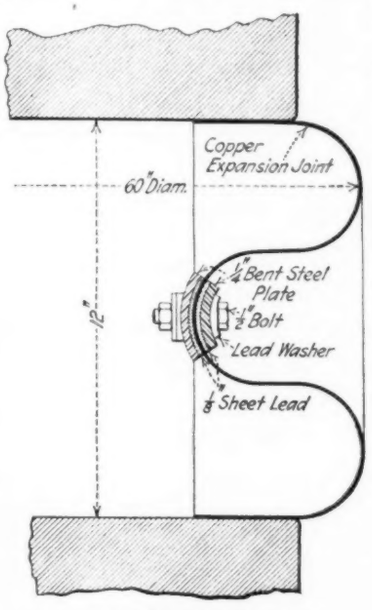
It is the writer's belief that the load conditions in many cases which have been presented to the public have been so chosen that the power rates derived might seem at first glance usable. It can be readily seen, however, that under many load conditions these would be far from workable. It is also but fair to state that in some cases unjust and extremely unfair rates have been given to companies whose stockholders were also interested in the central station giving the rate. This, however, is largely a thing of the past.

F. TARR.

Berkeley, Calif.

Air Leak in Condenser System

In the issue of Oct. 19, page 558, Francis McGough, Jr., tells of his troubles in locating air leaks in the condenser piping. This is a comparatively common trouble with the copper expansion joint, and few large ones get by without giving more or less trouble from small cracks in the corrugations. One very effective way to stop the extension of these cracks and also to stop the leak is to drill a hole through the center of the crack (say $\frac{17}{32}$ in.) and put a soft patch on it. If this is not possible, the joint must be taken out and brazed, and at the same time the whole joint should be annealed, which will add considerably to its life.



EXPANSION JOINT CLAMP

To properly test a condenser and piping, fill them with water; if there are any leaks, they will show up and the proper remedy can be applied. If only a drop of water leaks out, it should be stopped at once, for a very small air leak will seriously affect the vacuum. Leaks can often be stopped by painting over the exterior surface with heavy asphalt paint while the vacuum is on the system; the external air pressure will force the paint into the smallest holes.

FREDERICK L. RAY.

Louisville, Ky.

Surface-Condenser Location

In the issue of Oct. 26, p. 591, Edward T. Binns suggests a discussion as to the best place to put a condenser—whether in the supply or in the discharge lines of pumping units. A great deal depends upon its design, the conditions under which it has to work and the space available. The condenser shown in Mr. Binn's letter would work better if the suction head was above atmospheric pressure or if it was placed on the discharge side of the pump.

Both locations have their advantages and disadvantages. If the condenser is placed on the suction side it is generally in such a place that it is difficult to get at to repair and clean and therefore is often neglected. It is at a great disadvantage when starting the pump, or if the pump should lose its suction water—if there is no other means to supply the condenser with water. An advantage is that should the tubes leak the leakage would not be as great because of the low pressures.

The advantage of having the condenser on the discharge side is that it is generally more accessible, easier to repair and clean, hence more likely to be kept in good condition. When starting the pump the condenser is full of water and a vacuum can be obtained. The disadvantage is that should the tubes leak the leakage would be more, as the range of pressure on the cooling-water side would be greater.

I would prefer to have the condenser on the discharge side for low discharge pressure with considerable suction lift, and on the suction side for high discharge pressure where the suction lift is low or flows in by gravity.

Kansas City, Mo. LAWRENCE KJERULFF.

Engineers' License Laws

At present the engineers' license laws do not take into consideration the man who has the final say in the things vital to the welfare and even the safety of the power plant—namely, the superintendent, manager or whatever title he may have. In many establishments the chief engineer is the only man who can pass an examination and really knows the requirements of the power plant, yet another, perhaps a clerk or disbursing officer, may influence or even block the safety plans worked out by him.

A discussion, in *Power,* of this phase of the subject may prove beneficial. A. C. RUHL.

Port Clinton, Ohio.

Triplex Boiler-Feed Pumps

The question was raised some time ago, "Why are centrifugal pumps not making greater headway for boiler feeding?" Another equally important question is, Why are triplex power pumps not being used more for boiler feeding? They have most of the advantages of the duplex steam pump and few disadvantages.

Centrifugal pumps are not efficient as boiler feeders when used in plants of less than about 1,000 hp., and then they must be operated at nearly rated capacity to be at all efficient, and even then their efficiency, except under most favorable conditions, is only about 50 per cent. The triplex pump is efficient in plants of 500 hp. or less and may be driven by a variable-speed electric motor if the electric supply is reliable and continuous, or by a small high-speed engine using about 50 lb. of steam per horsepower-hour. Duplex pumps of small sizes have a steam consumption of from 100 to 200 lb. per hp.-hr.

Where stokers of the forced-draft type are used, the engine that drives the fan may be used to drive the triplex pump also. The feed pump would then be operated at a speed in proportion to the amount of steam used and would need very little other regulation. If automatic feed-water regulators are used it is necessary to have a relief valve set about 30 lb. in excess of the boiler pressure in the discharge line (probably bypassed back to the suction) to eliminate danger of overpressure.

The triplex pump is simple, gives a nearly constant flow of water, is about equally efficient at all speeds, ranging from 70 to 85 per cent. The first cost of a pump and motor, however, would be higher than that of a duplex pump.

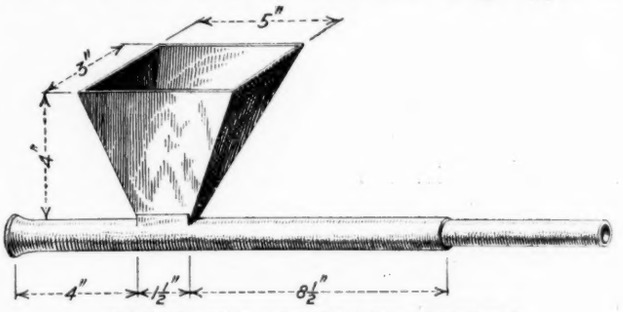
It seems to me that the use of geared triplex pumps to displace duplex steam pumps in plants that are too small to make use of centrifugal pumps would prove decidedly advantageous. J. C. HAWKINS.

Hyattsville, Md.

Cement Gun for Boiler Settings

In the May 18 issue of *Power* I noticed a description of a gun for squirting cement into the cracks of boiler walls, by George C. Abbe. In going over our boiler settings we successfully used a similar gun, but of somewhat different design.

From a short length of brass tubing about an inch in diameter was sawed a piece 14 in. long. Four inches from the end an opening 7/8 in. wide and 1½ in. long was cut with a sharp chisel, the tube being slipped on



CEMENT GUN FOR BOILER SETTINGS

to an iron rod held in the vise, while cutting. The cut was made along the center of the opening, and the metal was bent out straight along the two sides. To the two sides was attached a hopper, 4 in. deep, and 3 by 5 in. at the top, made of tin. The short end of the tube was given a flare of about a quarter of an inch, and a short piece of loosely fitting broom handle for a plunger completed the gun.

The plunger should not be too snug a fit, and the sand should be screened through a fine screen, or trouble will be experienced from sticking of the plunger. By rinsing the apparatus in a bucket of water each time before loading, the cement will flow more freely.

East Las Vegas, N. M. HARRY ROEBUCK.

Keeping Oil Warm in Winter

Sometimes it is the small things that help to make the wheels of industry run smoothly. My cylinder-oil tank is in a cold room, and the oil is consequently hard to handle. I started to put in a steam coil to keep it warm, but that required considerable piping, so I changed my plan and used a lamp immersed in the oil instead.

I used cotton-wrapped lead-covered wires, a porcelain socket and a weight to submerge the lamp in the oil. The temperature can be regulated by the size of the lamp to keep the oil just right to handle. I use a 16-cp. carbon-filament lamp. E. J. SAXE.

Chicago, Ill.

Inquiries of General Interest

Low Boiler Efficiency with High Percentage of CO_2—In boiler-furnace operation is it possible to obtain a high percentage of CO_2 and still be operating at a low percentage of efficiency, and if so, under what conditions? E. L. W.

High percentage of CO_2 might be representative of the quality of combustion of so small quantity of coal that the radiation losses from the boiler and setting would be so large a proportion of the heat developed by the combustion that there would be a low percentage of boiler efficiency due to the small percentage of heat available for the boiler. The conditions might obtain, using a large boiler to supply a very small demand for steam.

Safe Load for Piles—What is the safe load for piles which, when driven under a 2,500-lb. hammer falling 15 ft., make a penetration of 1 in.? W. F.

The safe load in tons per pile is given by the formula

$$2\,Wh \div (S + 1)$$

where

W = Weight of hammer in tons;
h = Height of fall of the hammer in feet;
S = Penetration from the last blow in inches.

Substituting the values given, the formula for safe load per pile becomes

$$2 \times 1.25 \times 15 \div (1 + 1) = 18.75 \text{ tons}$$

White or Snowy Center in Can Ice—What is the cause and remedy for a white or snowy center forming in can ice? J. R. R.

More or less of such appearance occurs in all can ice and is more pronounced in ice made from raw water or water containing impurities, either in the form of air or other gases that are liberated out of the water itself or out of the impurities, just before the final closing up of the ice in the center of the can. Various remedies are recommended for this difficulty, one of the simplest being to stir the water or agitate it with compressed air near the surface during freezing. Another is to pump out the water left in the can just prior to the time when the center freezes and replace the void with distilled water.

Economy of High Boiler Setting—We have three 72-in. by 18-ft. return-tubular boilers that are fired with soft coal and are forced beyond their capacity. The boilers are set 5 ft. 6 in. above the grates. Are not the boilers set too high for best economy? L. H. B.

The height above the grates is rather more than usually employed, but is advantageous in securing greater economy by obtaining more perfect combustion and higher temperature of the fire. These should more than compensate for the reduction in intensity of radiant heat due to having the boilers set higher than usual above the fires; and even though the front ends of the boiler shells may receive no more heat by direct radiation, there should be a higher temperature of furnace gases available for the tubes and other heating surfaces beyond the bridge-walls.

Loss of Draft in Smoke Flues—How much force of chimney draft is lost from friction due to length of flues and turns at elbows? R. C. M.

For average sizes of flues, the losses are about 0.1 in. (water pressure) force of draft per 100 ft. length of flue, and for each right-angled turn there is a loss of about 0.05 in. Where turns are made to a radius of five to six times the diameter of the flue, the loss is practically no more than that due to an equal length of flue. For a stack temperature of 450 deg. F. and external air temperature of 80 deg. F., the draft in a 100-ft. stack would be about 0.55 in., and the draft per foot of height would be practically 0.0055 in. It is seen, therefore, that for a given draft each right-angled turn in the breeching will require about 10 ft. additional height of stack, and each additional foot in length of flue will require about 2 in. additional height of stack.

Boilers Leak From Unequal Expansion—We have three 72-in. by 18-ft. return-tubular boilers that are fired with waste strips, sawdust and shavings, and have experienced considerable trouble from leaky tubes, seams and rivets. It is our custom to hold steam in one of the boilers day and night and to restart fires under two boilers daily at 4 a.m., taking two to three hours for raising steam. Would the daily restarting of the boilers be a greater cause of leakage than if steam was held in all boilers for 24 hr.? A. W. D.

The use of light wood, sawdust and chips for fuel gradually and evenly fired for restarting the boilers in two to three hours should not injure them materially more than ordinary firing with the same kind of fuel during 24 hr. Unless provision is made for continuous firing and a uniform supply of air (both of which would be difficult of attainment with the kind of fuel used), there is great likelihood of frequent and sudden changes of furnace temperature from admission of cold air through the fire-doors, resulting in leaks caused by unequal expansion.

Current Required by Induction Motor—The power factor of a 500-volt three-phase induction motor is 0.85, and at full load the motor takes 25 hp. from the line. What will be the full-load current in each line wire? B. L. R.

The input may be expressed by the formula

$$\text{Watts input} = E \times I \times 1.71 \times \text{power factor}$$

where E is the voltage across any phase and I is the current in any lead. Since one horsepower is equivalent to 746 watts, the horsepower input may be expressed as

$$\text{Horsepower input} = \frac{E \times I \times 1.71 \times \text{power factor}}{746}$$

Substituting,

$$25 = \frac{500 \times I \times 1.71 \times 0.85}{746}$$

whence

$$I = \frac{25 \times 746}{500 \times 1.71 \times 0.85} = 25.66 \text{ amp.}$$

Coal Cost for Operating Feed Pump—What is the cost of pumping 1,000 lb. of feed water with a duplex boiler-feed pump to supply boilers operated at 100 lb. gage pressure? C. V. H.

The factors of expense are variable with conditions. Assuming that the water temperature is 62 deg. F., its density would be 62.355 lb. per cu.ft., and for delivery of 1,000 lb. of water there would be a pumpage of $\frac{1{,}000 \times 1{,}728}{62.355}$ cu.in. If there is a loss of 10 lb. per sq.in. for opening the discharge valves and for overcoming friction of the pump-discharge passages, elevation of the water and friction of the pipe line and fittings and also a boiler pressure of 100 lb. per sq.in., or 115 lb. absolute, then the pressure on the discharge side of the water piston would have to be 125 lb. per sq.in. absolute. If there is a water-cylinder pressure on the suction side of the piston of 20 lb. per sq.in. above atmospheric pressure, or 35 lb. absolute, the net water-piston pressure required for performing the work would be $125 - 35 = 90$ lb. per sq.in. With the same net pressure and regardless of the size of pump, the work performed per cubic inch of pumpage would be the same as for a piston having 1 sq.in. area and 1-in. stroke, namely, $1 \times 90 \div 12$ ft.-lb., and therefore the actual pumpage would require a development in the water cylinder of $\left(\frac{1{,}000 \times 1{,}728}{62.355} \times \frac{90}{12}\right)$ ft.-lb. Allowing that 20 per cent. of the initial power developed is lost in overcoming friction and inertia of moving parts, then the gross work developed in the steam cylinder would need to be $\frac{1{,}000 \times 1{,}728 \times 90}{62.355 \times 12 \times (1.00 - 0.20)} = 259{,}802$ ft.-lb.

Assuming a consumption of 85 lb. of steam per hour per indicated horsepower (or per $33{,}000 \times 60$ ft.-lb.), that 1 lb. of coal is required for generation of 8 lb. (weight) of steam, and that the cost of coal is $3 per ton of 2,000 lb., then the cost of work developed by the steam end of the pump would be

$$\frac{85}{33{,}000 \times 60} \times \frac{1}{8} \times \frac{\$3.00}{2{,}000} = 0.00000000805 \text{ per ft.-lb.}$$

and under the conditions assumed the cost of coal for operation of the feed pump would be

$$259{,}802 \times \$0.00000000805 = \$0.00209 \text{ per } 1{,}000 \text{ lb.}$$

of boiler-feed water used.

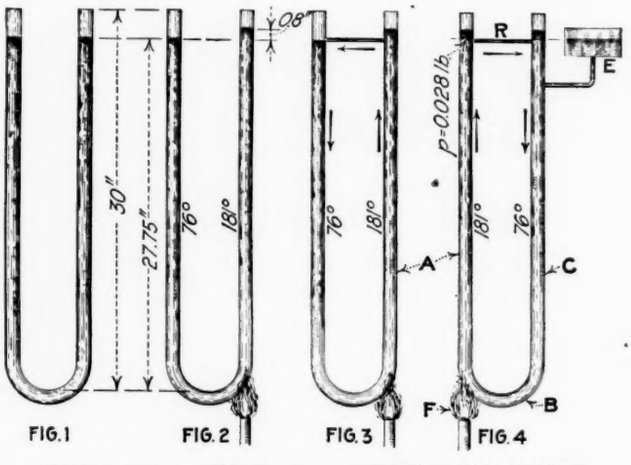
Elements of Hot-Water Heating*

By George E. Reid

SYNOPSIS—This paper explains the principles underlying the operation of a hot-water heating apparatus and also considers some practical methods used in designing the piping system.

The operating principle of a hot-water heating system may be best understood by considering an open vessel containing water in which are floating particles of some dark substance. If the water is heated, ascending or descending currents are made visible. This movement is called circulation and is typical of what goes on in the piping of the hot-water system. What happens is that the water that comes in contact with the heated surface expands. It becomes lighter and is floated to the top of the vessel by the surrounding cooler water. The latter sinks to the bottom, displacing the hotter water, and is in turn heated, producing the current or circulation previously mentioned. This circulation soon equalizes the temperature by diffusing heat throughout the mass of water. The action is called "convection," hence the name "convection water" applied to the water circulated in a heating apparatus.

For an illustration of the circulating force, imagine a glass tube of U-shape, as shown in Fig. 1. This is of considerable length, of uniform bore and is partly filled with cold water. As there is an opening between the two legs at the bottom of the U, the water will stand at the same height in both and will be in equilibrium. Now if a spirit lamp or other source of heat be applied, as in Fig. 2, to the bottom of one leg, the water in that leg will become heated and will perceptibly expand. The top of the water in the heated leg will stand higher than that in the cold leg, and yet both columns of water are the same weight. This of course is caused



FIGS. 1 TO 4. HEAT AS A CIRCULATING FORCE

by the expansion of the water and illustrates the well-known law of physics that the volume of water for a given weight, or the specific gravity, decreases with the increase of temperature.

Suppose the U-tube is made 30 in. long and filled with water at 76 deg. F. to a point 27.75 in. above the top of the water in the lower bend (Fig. 3). Let heat then be applied to one leg until the water therein attains a temperature of 181 deg. The surface of the water in the heated leg will then be about 0.8 in. higher than in the cold leg, provided the temperature of the latter has remained stationary. Now imagine that the upper ends of the tube are connected by a small tube at the level of the top of the water in the cold leg (heat being still applied), as shown in Fig. 3. Since water will flow from a higher to a lower level, a flow will occur from the hot to the cold leg until both temperature and level are equalized, when the circulation will stop.

But if the water could be cooled while passing through the connection pipe at the top, the circulation would recommence and continue as long as heat was applied to one of the columns. Go a step further and imagine that the upper ends of the U-tube are plugged at a height of 27.75 in. It was seen (Figs. 2 and 3) that when heat was applied to one leg there was an expansion of the water. To prevent this expansion from acting destructively on the closed U-tube, suppose a tube be run from the cold leg to a small open reservoir, or expansion tank, placed at such a level that the surface of the water shall be at the 27.75-in. level. The tube with such an expansion tank is shown in Fig. 4. If heat is now applied the expansion of the water in the hot leg, instead of raising the level, will force some water from the cold leg into the expansion tank. Both legs are plugged and the water must stand at the same height in both. As the hot column is lighter height for height than the cold column, there will develop through the water a gradually increasing pressure that will result in a flow from the hot leg through the small connecting tube into the cold leg. If the water is cooled in passing through this connection, as before, and the temperature of the hot leg reaches the former temperature of 181 deg., a pressure of approximately 0.028 lb. per sq.in., equivalent to 0.8 in. of water column, will be developed at the top of the hot leg. The water will flow as rapidly as when there was a difference in level between the open legs.

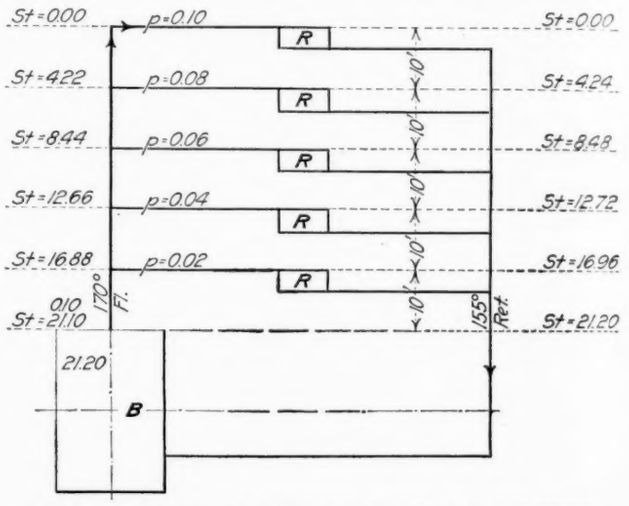


FIG. 5. FORCE OF TEMPERATURE DIFFERENCE

MINIATURE HOT-WATER HEATING SYSTEM

The essential operation of a hot-water heating system is represented by the conditions indicated in Fig. 4. The U-tube is a miniature system in which the hot leg H is the so-called flow riser, the cold leg C the return riser, the connection R at the top between the legs the radiators, the open reservoir E is the expansion tank, the bottom B of the hot leg is the boiler, and the flame F is the furnace.

It has been shown that with a difference of 105 deg. temperature and a column 27.75 in. high, a certain pressure difference is maintained, and it also follows that there will be a pressure difference between the flow and return risers for any difference in temperature or any height of column. A positive downward pressure exists at the top of the hot leg. This pressure exists through the mass of that column down to the boiler or lowest horizontal distributing main. The pressure difference between the hot and the cold leg will vary with the height above the boiler, for at the top of the cold leg the static head, or pressure, is zero, and consequently at that level the pressure difference is at its maximum. At the bottom of the U-tube the pressure difference is zero, for at that level the forces are in equilibrium.

Fig. 5 demonstrates the magnitude of the forces available at different levels. B represents the boiler, FL the flow riser, R the radiators, RET the return riser, P the pressure difference between risers, and ST is the static pressure, both in pounds per sq.in. It is assumed that an expansion tank is placed at such a height that its water level is at the top of the highest radiator, though the static head will be zero at that level. The temperature of the water in the flow riser is assumed to be 170 deg. and in the return riser 155 deg., a drop in temperature in the radiators of 15 deg.

*From a paper read before Washington No. 1, National Association of Stationary Engineers, Washington, D. C.

The weight of the water in both legs is of course pressing downward. The static pressure at the top of the boiler, or at the base of the floor riser, is 21.10 lb. per sq.in., while the static pressure at the same level in the return riser is 21.20 lb. per sq.in. The heavier column of water in the return riser forces the lighter column in the flow main upward, and in this case with a force equal to the difference between the static pressure in the two columns, or 21.20 — 21.10 = 0.10 lb. per sq.in., equivalent to a height of water column of 2.97 in.

The pressure difference at the 50-ft. elevation is 0.10 lb. per sq.in. If this is added to the static head in the flow main at the 40-ft. level and the static head in the return riser at that level is subtracted from the sum, the result will be the pressure difference beween the flow and return risers at the 40-ft. level. For example, 0.10 + 4.22 — 4.24 = 0.08 lb., or 2.374 in. of water column. This is the difference between the static pressure at the base of flow and return risers 40 ft. high, or 16.96 — 16.88 = 0.08 lb., thus demonstrating that the pressure difference at any level is equal to the difference between the weight of two equal columns of water at different temperatures. Fig. 5 can be similarly used to find the pressure difference at any 10-ft. level below 50 ft.

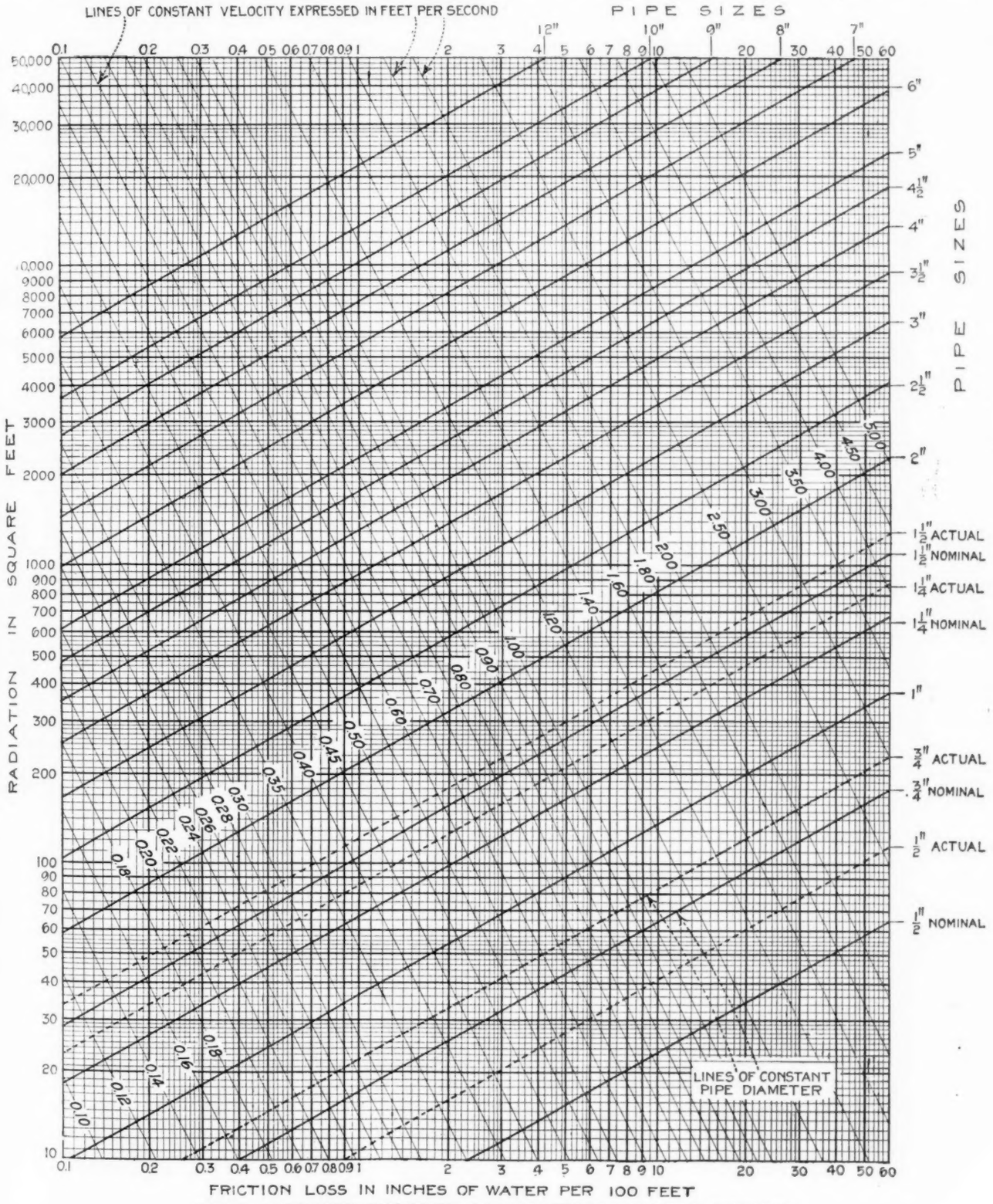


FIG. 6. PIPE SIZES REQUIRED FOR HOT-WATER HEATING SYSTEMS

As the pressure differences increase with the height of the building, it is necessary to devise means for controlling the convection water so that it may be equally distributed at all levels. This is sometimes done with specially graduated radiator valves operated by hand. The same result is more satisfactorily obtained by proportioning the risers so that the flow of water may be controlled by pipe friction. It is also possible to equalize the friction in the mains delivering the convection water to the risers, and in connection with the riser proportions to design an apparatus so that the entire system may be equally efficient or as near so as the commercial pipe sizes will allow.

The accompanying logarithmic chart has been constructed for the purpose of readily determining the pipe sizes and solving other problems arising with hot-water heating installation. The pressures in the chart are expressed in inches of water rather than in pounds per square inch, as the differences are thus more apparent. The horizontal lines across the chart, with the figures at the left-hand margin, denote various amounts of radiation in square feet. The vertical lines with the figures at the upper and lower margins denote pipe friction in inches of water per 100 ft. of "water travel."

TABLE 1. LENGTHS (INCHES) TO BE ADDED FOR VALVE FITTINGS

Size of Pipe	Elbows	Globe Valve	Size of Pipe	Elbows	Globe Valve
½	2	5	4	13	26
¾	2½	6	4½	15	30
1	3	7	5	17	34
1¼	4	8	6	20	40
1½	5	10	7	23	46
2	7	14	8	27	54
2½	8	16	9	30	60
3	10	20	10	33	66
3½	12	24	12	36	72

The diagonal lines rising from left to right with the figures on the right-hand and top margins indicate the pipe diameters in inches. The diagonal lines rising from right to left are velocity lines, the figures on which indicate the velocity of water travel in the pipes in feet per sec.

USE OF CHART TO FIND PIPE SIZES

In order to use this chart first find the total length of water travel; that is, the length of flow and return pipes plus an allowance (see Table 1) for valve fittings. The system should be so designed that the flow and return water travel in the same direction in the horizontal runs. The convection water then travels the same distance to and from any set of risers.

TABLE 2. HEAD FOR VARIOUS ELEVATIONS

Elevation in Ft.	Head, In. of Water	Elevation in Ft.	Head, In. of Water	Elevation in Ft.	Head, In. of Water
5	0.30	50	2.97	95	5.64
10	0.59	55	3.26	100	5.94
15	0.89	60	3.56	105	6.23
20	1.19	65	3.85	110	6.53
25	1.48	70	4.15	115	6.83
30	1.78	75	4.45	120	7.12
35	2.08	80	4.75	125	7.42
40	2.37	85	5.05	130	7.72
45	2.67	90	5.34	135	8.01

To find the sizes of the horizontal mains and their branches, first compute the friction loss per 100 ft. This is done as follows: From Table 2 find the head in inches of water corresponding to the height of the highest radiator above the boiler; from this subtract the velocity head, which may be assumed to be 0.3 in. Divide the remainder by the total water travel in hundreds of feet, and the result is the friction loss per 100 ft. The pipe size is found by entering the chart from the left at the proper amount of radiation and following the horizontal line to its intersection with the vertical line for the previously determined friction loss. From this intersection follow the nearest diagonal pipe line to the right-hand or top margin, where the desired size will be found.

As an example, suppose it is required to find the size of a horizontal main or branch needed to feed 700 ft. of radiation in a system where the total water travel is 1,000 ft. and the top of the highest radiator is 40 ft. above the boiler. From Table 2, at a 40-ft. elevation, the head is 2.37 in. of water; then

$$\frac{2.37 - 0.30}{10} = 0.207,$$ say, 0.2 of an inch, which is the allowable friction loss per 100 ft. Enter the chart at the 700-ft. radiation line and follow along horizontally to its intersection with the 0.2 vertical friction line. The nearest pipe line is the 4½-in., the size desired. The nearest velocity line is the 0.3, showing that when a 4½-in. pipe is heating 700 ft. of radiation the velocity of the water will be 0.3 ft. per sec.

To find the riser size or capacity for an upfeed one- or two-pipe or for a downfeed two-pipe system, enter the chart with the given radiation or pipe size but with a friction loss per 100 ft., as shown in Table 3. For one-pipe downfeed, use the value in Table 3 for one-half the height for both the main riser and drop riser sizes.

TABLE 3. RISER FRICTION LOSS PER 100 FT.

Elevation in Ft.	Head Lost, In. of Water	Elevation in Ft.	Head Lost, In. of Water	Elevation in Ft.	Head Lost, In. of Water
5	0.10	40	2.34	75	2.65
10	0.47	45	2.41	80	2.65
15	1.30	50	2.47	85	2.67
20	1.71	55	2.51	90	2.69
25	1.97	60	2.55	95	2.70
30	2.14	65	2.58	100	2.71
35	2.25	70	2.60		

By the use of these tables and the chart the size of the main riser for a two-pipe downfeed system to feed 2,100 ft. of radiation when the top radiator is 40 ft. above the boiler is found to be 4 in.; the size of the main riser for a single-pipe downfeed system for a 40-ft. elevation and 2,100 ft. of radiation is 4½ in.; and of a drop riser to feed 40-ft. radiators with 300 ft. of radiation is 2 in.

TEMPERATURE CORRECTION OF CHART RESULTS

These computations are based on a temperature drop of 15 deg. in the convection water in the circuit and an average temperature of 163 deg. If it is desired to proportion the piping for other temperature drops proceed as follows: For 20 deg. drop enter the chart at three-quarters of the desired radiation and take 1⅓ times the head caused by the 15-deg. drop. At 30-deg. drop enter the chart at one-half of the desired radiation and take twice the head caused by the 15-deg. drop.

Pipe friction, or loss in pressure (head), is one of the principal factors entering into the solution of the examples cited. Friction is always present when the water is in motion and must be recognized and equalized when proportioning the apparatus if it is desired that all radiators operate with equal efficiency.

The chart may also be used to obtain the size of the radiator tappings and to determine the pipe sizes for forced circulation, either by velocity or friction, since its range is wide enough for all practical purposes.

The World's Supply of Fuel and Motive Power*

Quoting from the first Census of Production (1907) reciprocating steam engines in use were rated in round numbers at 8,000,000 hp., steam turbines at 530,000, internal-combustion engines at 680,000 and water-power engines at 178,000, and also in round numbers there was one horsepower per person employed in manufacturing industries in the United Kingdom. The quantity of fuel consumed in providing this amount of power is naturally enormous, but it is satisfactory to know that if the demand is not increased—a big if of course—the world's supply of coal would last for 5,400 years, and that in the United Kingdom would last at the present rate of consumption, 500 years. Nearly 27 per cent. of all the coal burned in Great Britain was used in factories, and it is claimed that 30 per cent. of it could be saved by gas generators and engines; 21.4 per cent. was used for domestic purposes, and 8 per cent. of this, it is estimated, could be saved by gas cooking and heating with briquettes and coke. Taking all the lines of economy, it is estimated that about 60 per cent. of the coal burned annually might be saved.

The speaker suggested some lines improvement might take. For example, more use might be made of the exhaust heat from steam and internal-combustion engines; large central stations with gas generators and high efficiency gas engines would reduce the consumption of power for factories to one-third of its present amount; and steam heating and cooking would enable the heat of 10,000,000 tons of coal to do the domestic work now performed by 36,000,000 tons. On the whole, the lecturer came to the conclusion that half the present consumption of coal could be saved. He touched briefly upon solar energy, but devoted more time to water power. "With increasing scarcity of coal," he said, "undoubtedly hydraulic power will in the future show greater advantages, and even in England and Ireland it might be possible to earn interest on capital expenditure greater than $200 per hp." He estimated that by great engineering works, it might be possible to obtain perhaps 3,000,000 hp. from areas that could be given up for the purpose. "Altogether, to carry on the industrial civilization of these islands in time of peace, on the scale of today, absorbs about 19,000,000 hp."

*A brief review by the "Engineer" (London) of a lecture by Dugald Clerk before the Institution of Mechanical Engineers

it is evident, then, that it is the engineer's business to economize coal. In a comparison of the steam and internal-combustion engine he showed that motors of the latter type may give an indicated thermal efficiency of as much as 40 per cent., which is much higher than that given by a steam plant. It was hoped at one time that gas engines could be indefinitely enlarged, but experience has shown that that is impossible, and everything tends to restrict the commercially salable internal-combustion engine to moderate powers. Closing, he said: "So far as I can see, there is no hope of an indefinite increase in power using reciprocating pistons. Something must be done to introduce the rotary principle." He suggested that by the combination—which, we believe, was foreseen by Mr. Humphrey and by other inventors—of a pump of the Humphrey type and a Pelton wheel, some advance might be made, and with such mechanisms built in large units that the power needed by Great Britain could be produced for one-third the amount of coal now used.

The thermal efficiency of the heat engine has advanced from 3.8 in 1820 (Boulton and Watt) to 23 per cent. in the Parsons turbine and 40 per cent. in the Diesel engine. "Meantime, by the application of high-efficiency engines, the use of all waste heat for domestic and industrial purposes and the application of all available water power, all on a large scale, the engineer may extend the industrial period in England to over one thousand years."

The Lubrication of Bearings and Cylinders*

By F. L. Fairbanks†

SYNOPSIS—Micrometers show that the oil film under a shaft gets thicker as the speed of the shaft increases, the journal acting as an oil pump. Oil grooves are not approved unless the edges are well rounded. Pistons are made to ride on a film of oil by grooving the cylinder bore; the piston also is grooved, or rather threaded. As viscosity and gravity of system oils decrease due to "wear" and as shown by daily reports, they are increased by the addition of a "strengthening" oil. Viscosities of supposedly like oils vary at high working temperatures.

What I shall say is about the experience I have had and about the deductions based on that experience. I do not intend to go into the technical side of lubrication. There is almost no bearing condition that cannot be readily corrected unless so bad that the machine is perhaps gone beyond the possibility of recovery. Plants run satisfactorily for weeks, and then some day the operating force reports a warm bearing, and within an hour or two there is another, and sometimes half of the station seems to be in trouble within two or three hours. Naturally, the trouble is laid to dirty oil, poor filtering, etc. My experience has demonstrated that such action is a simple matter in physics and can be easily controlled if the correct methods and the right materials are used.

THICKNESS OF OIL FILM IN BEARING

Another thing that appealed to me was the question of so machining bearings and cylinders that the handwork, so common and so costly in heavy-engine building, could be eliminated. And beginning with the work, I wanted to find out what the probable thickness of the film of oil was under a joint, under a piston, or under whatever might be lubricated by the oil.

My first experiments were to put a dial micrometer on the main-bearing caps of one of the large machines, drop the spindle until the shaft touched the bearing and take the readings at various speeds and with the engine standing still. Experimenting showed that under normal running conditions the average film of oil under the shaft was from 0.002 to 0.003 in. and that the thickness of the film depended considerably upon the rotative or circumferential speed, increasing with the speed. This is undoubtedly because more oil is pumped or pulled in between the journal and the bearing as the speed increases. When the engine is shut down and allowed to stand for half an hour, metal touches metal; in starting, the dial micrometer indicates the rise of the shafting.

As a result of this experimenting we give the average journal 0.005 in clearance and, to eliminate hand work (we have a four-piece bearing), machine the bearings 0.005 large and take out the shims that are placed between the cheek pieces and the upper and lower shell after the machining. That gives a shape of bearing that is almost identical with the hand-scraped bearing.

It is the custom with most engine builders to bore the shell practically the exact diameter of the journal and then scrape the sides of the shell sufficiently to allow the oil to work in, in the judgment of the man who scrapes it; that is, practically all erectors or engine builders cut oil grooves in the bearing to carry the oil under the journal.

I have used several babbitted bearings that ran warm; in fact, they started the babbitt and filled the oil grooves and then ran cool. I found that the average oil groove, as cut, is of comparatively little value because it is usually a channel with square edges, and if the journal does not lift on the film of oil, the oil cannot get over the sides of the channel.

Working out this problem, it seemed to me something like this: If a heavy weight is upon skids being carried upon

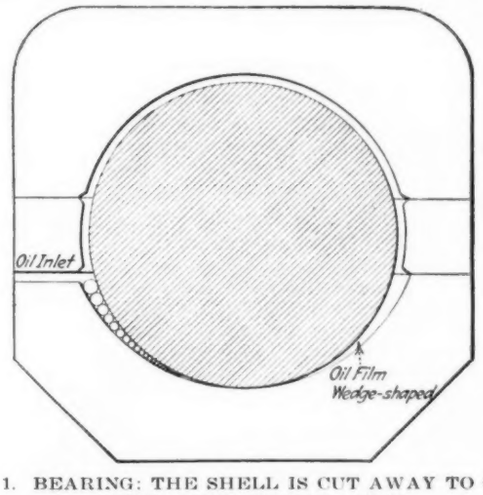


FIG. 1. BEARING: THE SHELL IS CUT AWAY TO GIVE A WEDGE-SHAPED OIL FILM

rollers, one would not think of using a rectangular-section skid to pick up each successive roller. Almost any laborer knows enough to round off the under corner of the skid so that it will easily go on the roller. The sag of the skid due to the load is picked up gradually, lifting the skid.

JOURNAL ACTS AS AN OIL PUMP

That is what takes place in a bearing. The journal, in revolving, picks up the oil and carries it down under the shaft, raising the latter sufficiently to allow it to ride on a film of oil. From these deductions I started by boring the shells in the manner described and as shown in Fig. 1. We cut off 1 to 1½ in. at the bottom to get rid of that corner at the edge: this gives a wedge-shaped opening from either side down. The back side does little good when cut away because the oil is coming out there, but to machine rapidly one must machine both sides.

The theory is that the oil is fed in and is dragged down under the shaft supporting it. This action depends considerably upon what may be called the "tensile strength" of the oil, or its viscosity.

*Lecture at the Worcester Polytechnic Institute, Worcester, Mass.

†Chief Engineer, Quincy Market Cold Storage and Warehouse Company, Boston, Mass.

The first experiment was performed with two oils approximately of the same tensile strength and viscosity, and a different result was had with each. Animal oil produced a different result from mineral oil of approximately the same viscosity. This supports the conclusion that there are two qualities that represent the lubricating properties of the oil. The first is viscosity, or tensile strength, the second is adhesiveness to the metal surfaces. The tendency for the animal oil to adhere to the surfaces increased its value because, instead of slipping on the surfaces, the increased adhesiveness helped carry it under the shaft; a dial micrometer showed a rise and fall of the shaft as the latter's speed increased and decreased—the shaft, owing to the construction of the bearing, becomes an oil pump, dragging oil in, the oil compressing, giving way or elongating. It is only necessary for the oil to maintain its film on the bottom of the shell to lubricate properly.

ADVANTAGES OF THE RUSSIAN OIL

A number of mixtures of mineral, animal and vegetable oils were tried and it was found that the vegetable oil did not take kindly to heating; the viscosity of that oil—castor, olive, etc.—was higher as a rule, but as they heated they lost viscosity rapidly—much more rapidly than the mineral or animal oils. Animal oils, such as tallow and lard, beyond a certain point lost their viscosity faster than the mineral oils. We experimented with the three classes of mineral oils—the

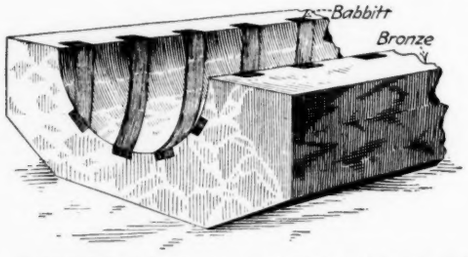


FIG. 2. BRONZE SHELL WITH DIAGONALS OF BABBITT

asphalt, the paraffin and the olefine. Practically all American oils come from either the asphalt or the paraffin. Russian oils from the Caspian Sea and Galicia are of a different character, being of the olefine base.

We imported some Russian oil, experimented and found that the "tensile strength," or viscosity, was always in excess of the paraffin or asphalt oil, the asphalt being the lowest. The Russian oil carries a much lower cold test, more viscosity value and longer range than either of the others. As the cold-test temperature goes down with the paraffin or American oils the flashpoint also drops, but the olefine, or Russian, oil goes farther in both directions than the American oils. There is less of the volatile element, and consequently the oil carries through a longer range without a change of its characteristics, as is customary with the paraffin, therefore we use the Russian oil.

This bearing construction as an aid to lubrication worked out so well that we apply it to all our engines. In keying an ordinary bearing, the first instruction we give to the men that do this is to allow 0.005 in. for the film of oil. We now design our engines with the wedge setscrews 6 threads to the inch, with a hexagon head and with a pitch to the wedge which gives 0.002 in. to the square of the head. When keying the different machines we have tables for guidance of the men which provide for the oil film and the expansion factor. So when a man is told to give a particular bearing 0.012 in., he drives the wedge "home" and backs off six squares, and is positive of what he is doing, and can key quickly.

ALLOWING CLEARANCE FOR EXPANSION

The expansion factor is based upon knowledge of what the approximate running temperature should be, when the temperature is dangerous, and at what temperature the engine would have to be stopped. Assuming the normal temperature of the bearing to be 85 or 90 deg. F., we conclude that when the bearing gets to be 175 deg. F. (that is about as high as a man can bear his hand on) it is time to stop the engine. It is simple, therefore, to calculate what clearance a journal should have. First, we give it 0.005 in. clearance for the oil; then we give it the expansion clearance. Figuring 75 deg. F. for the ordinary temperature and 175 for the maximum, we get a range of 100 deg. F. Multiply the 100 deg. by the coefficient of expansion to find what clearances to provide.

If a journal is 20 in. diameter, multiply the product of the coefficient of expansion and the range of temperature by the diameter of the journal; the result will be the distance to allow for clearance. Commonly, keying is accomplished by guess; the man doing the job guesses this or that will do and usually gives the 20-in. journal and the 10-in. journal the same clearance. If the temperature of the 20-in. journal increases as it is liable to, from a little dirt or some condition—and if the keying is inadequate, as the journal expands it soon reaches the point where it "pinches," and the moment the film of oil is all squeezed out the bearing begins to cut.

DIAGONAL BABBITT INSERTS IN BEARINGS

To further insure safety, all of our bearings now have a special bronze with diagonal babbitt inserts (Fig. 2). We use the babbitt as a lubricant and for its adhesive effect. We take a bottom shell and run diagonal strips of babbitt, the idea being that these strips will wear, usually (we find by measurement from 0.001 to 0.002 in.) under the surface of the bronze, as the oil apparently has a greater affinity for a good babbitt than it has for brass. The babbitt wearing from 0.001 to 0.002 in. under the surface of the bronze gives so many little oil channels that the oil globules will follow the journal. We found that this babbitt was one of the best safeguards we had. If an oiler was careless and allowed the bearing to run dry or with too little oil to carry the film around the shaft, these diagonal inserts of babbitt melted and immediately became a lubricant. Incidentally, they raise the shaft and protect it from the bronze bearing. The oiler usually will have a piece of waste in his hand and will take off the strainer to clean it and reach down to replace it. About two weeks ago, on No. 4 bearing of the 1,000-ton compressor[1] he evidently dropped a piece of waste into the hole before replacing the strainer. In about ten minutes, when again making his rounds, he saw a bright streak when he looked into that hole. He noticed that there was an increase in temperature at that streak and found it was "coming" (increasing), as we say. In three or four minutes he appreciated that it was gaining all the time. The machine was stopped by pushing a button, but before it stopped turning a film of babbitt about 1¼ in. wide had started and gone around the shaft. Upon examination it was found that the film had raised the shaft so that in the length of the bearing (40 in.) the journal had touched the bottom shell only on a strip 1¼ in. wide. That machine was shut down, and another was cut in. Ordinarily, the main bearing and the top shell and cheek pieces would have to come off. The main top bearing weighs 3,500 lb., and to cut that babbitt out would mean considerable work. All we did was to let up on the cheek pieces 0.003 in., and with the load off the engine, it was speeded up to about 60 r.p.m. This broke loose the film of babbitt, and as it came around we picked it off.

In the meantime, both ends of the journal were lubricated with a heavy oil. Picking the babbitt off left some dirt and rolled-up metal in the cheek pieces. We have a mixture of a fine grinding compound (quartz) known as "Trojan No. 1," which is much used for grinding brasswork. We mixed about one-third of the compound with one-third of the best lard oil and one-third of ordinary engine oil. By turning the shaft and dropping the compound across the affected part of the journal, the bearing was cleaned in less than an hour. All this time the bearing was being lubricated by an ordinary oil carried for that purpose—a heavy crude oil containing 25 per cent. of a high-grade lard oil. The machine was up to speed with a full load on in less than half an hour after the bearing was cleaned.

This shows what can be done to keep engines running. On the ordinary bearing, two things would happen: If it had been a full-babbitted bearing, it would have taken out from $\frac{1}{16}$ to $\frac{5}{16}$ in., which means that No. 4 bearing would have gone down $\frac{5}{16}$ in.; and with the rigid 20-in. shaft 26 ft. long, practically nothing could have prevented the cracking of the crankpin. Take $\frac{5}{16}$ or $\frac{1}{16}$ in. from under this bearing and something has got to happen to the center crankpin. The bronze insures a support under the bearing; the babbitt acts as a lubricant, and the raising of the shaft by the film of babbitt protects the rest of the journal even though it should cut where the film forms.

Attention to these little things has increased the efficiency of our plant and greatly lessened the fear of hot boxes.

One morning we had an oiler come on at 11 o'clock (the oilers are not allowed to make any changes in the method of oiling or the way of handling any apparatus). In this case, the oiler was "green" and had heard that an engine running constantly will get "gummed up." He thought it a good idea to put in kerosene and wash out the bearings, so in went a pint to the center crankpin bearing. At the time the compressor was working at 200-lb. pressure. Within a minute or so the engine began to slow down and the engineer stopped it; as it stopped there was quite an audible squeak. They then sent

[1]For description of this machine see "Power," Dec. 8, 15, 22 and 29, 1914, and Jan. 5, 1915.

for me, and I called in (as I usually do) by telephone all the men I thought I should use—and some more. Every bit of babbitt in that center crankpin box was wound round the shaft. We jacked up the piston in the cylinder and took off the end block, dropping the box. It was the worst-looking bearing I ever saw; if the box had been filled with gravel it could not have made a worse looking surface. Two men were put on with broad files, and in three hours the babbitt was off. There was hardly a scratch under the babbitt, which had cooled; there were a few streaks in the bronze bearings. In four hours the machine had its load. Emergencies such as

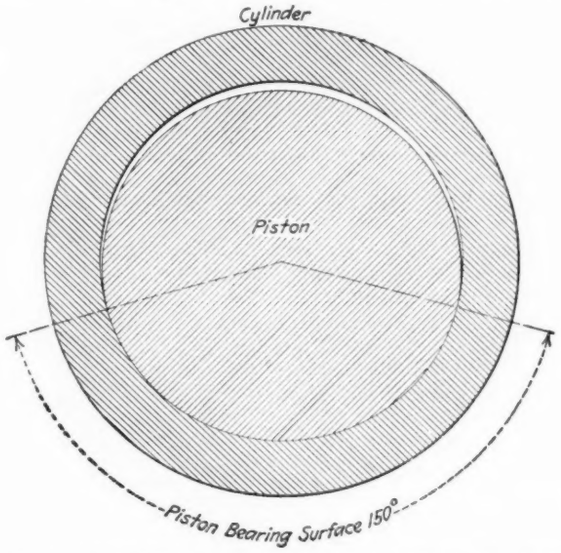


FIG. 3. PISTON HAS CONSIDERABLE BEARING AREA

this were provided for in the design, inasmuch as all of our bearings now carry a bronze surface; the one-third babbitt is extra, giving the added safety of the bronze shell to protect the journal. We can take out all of the babbitt and still have as much bearing surface as the average engine builder allows.

We have been running for more than a year with no babbitt in the low-pressure crank pin boxes and practically none in the lower half of the center crank-pin boxes. The foreman erector asked me, "What is the use of putting babbitt in there if you can run without?" The answer is that the babbitt is the best friction safety valve I know of.

CYLINDERS HAVE MANY OIL GROOVES

Our cylinder lubrication also is the result of research and experiment. We had a 60-in. low-pressure cylinder out of which I think we never got over three months' use. It carried a very heavy piston weighing three to five tons, with its portion of the rod.

The rod was dragging upon the stuffing-box and was ready for another setting up and either a new junk ring or new babbitt rings or composition rings were needed for the piston.

I had taken charge at this time and decided to eliminate the question of composition, alloyed metal, and use nothing but plain cast-iron junk rings. I visited a number of the larger shops and got all the information I could, and came back convinced that there was a radical improvement to be made in cylinder boring. The average engine builder carefully bores out cylinders, in fact some grind them to true; they then turn up the pistons with $\frac{1}{32}$ to $\frac{1}{16}$ in. clearance. This gives a circle within a circle which, as you know, must bear on a point; if it is a cylinder it bears on a line.

That is the custom in practically all the engine-building shops with which I am familiar. The result is that the builder has anywhere from a few pounds to five or six tons per square inch projected area in the cylinder, and the only reason that the engine can run is that the grain of the metal is close and the distortion of the cylinder is likely sufficient to increase slightly the bearing surface. There seemed no reason why we should not calculate the bearing surface the same as in bearings, and I immediately had a piston designed with the junk ring and we bored the cylinder without any particular care, anywhere within 0.006 to 0.010 in. Experience has shown me, especially in large cylinders, that every cylinder that is put into running condition is distorted. As it approaches the original temperature of the casting it is going back to the original shape. The distortion is proportional to the difference in temperature and the construction of the cylinder. When this is true, why waste time grinding cylinders?

NINE SQUARE FEET BEARING SURFACE IN CYLINDER

We turned the piston, which was 0.003 or 0.004 in. larger than the cylinder, jacked it off center and relieved it as in Fig. 3, giving the maximum of 150 deg. The old piston had a little over 7 sq.in. of bearing. With this method we had nearly 9 sq.ft.

It occurred to me that the average low-pressure cylinder must be running in water; that is, water between the piston and the cylinder.

There is a film of water anywhere from ½ to ¼ in. in depth (I have found as much as ½ in.) being pushed ahead of the piston. If this is true the oil for lubrication must be on the surface of the water and is not lubricating the point between the piston and the cylinder. The problem was to prevent this condition. I decided to bore out the cylinder, leaving the tool mark. It was bored out with 16 threads to the inch and looked like Fig. 4. We put in the 60-in. piston, and everybody thought it would make about two revolutions. That was nine years ago. Today these tool marks are black and the piston and cylinder do not touch, as the former runs on an oil film. The junk ring carries one ring of packing in the center as black as can be; where it overruns the packing ring the oil is gummed as it is on the back of a Corliss valve.

The piston rods on our ammonia machines are given 24 threads per inch where they pass over the packing, with the result that apparently the packing never touches the rod. The ammonia cylinders are threaded too.

GROOVES ACT LIKE A BALL RACE

In cylinder lubrication the buoyancy of the water does not seem to be sufficient to overcome the surface contact of the oil. I have been trying to account for this for some time and have several theories. On the pistons we have a broader tool-mark, which is shallow, so it will not measure the tool-mark on the cylinder. The ends of the piston are beveled. It seems to me that each of these grooves becomes a little oil compressor. As this bevel edge undertakes to push down the oil the resistance or reaction lifts the piston. Consider it another way: Each of the grooves acts as a ball race, the piston does not touch the cylinder. We can throw in a bucket of water and let it stand an hour or a day and draw off the water, and these grooves are still filled with oil. The only explanation I can give is that the adhesiveness of the oil to the surface is in excess of the gravity and therefore it does not rise to the surface of the water. By grooving the cylinder and allowing 120 deg. for contact, we found we could easily allow, under the existing temperatures, a pressure of 50 lb. per sq.in., but we do allow only 25 lb. We figure the flat tops between grooves as the projected area and subtract them from the net area, which gives several thousand times the projected area used by any other engine builder. This construction is somewhat expensive with the heavy pistons. We found, of course, that to get clearance we had to jack the piston three times. Now, instead of that we allow a close clearance. On a 64-in. cylinder we allowed 0.012 in. clearance; that is, we turned the piston 0.012 more than the cylinder. The aim is to get just clearance enough so the piston would clear; then the shallow grooves on the piston will cut until the piston gets the bearing it needs at the bottom. This works out well and does not cost any more than the old-fashioned way of turning.

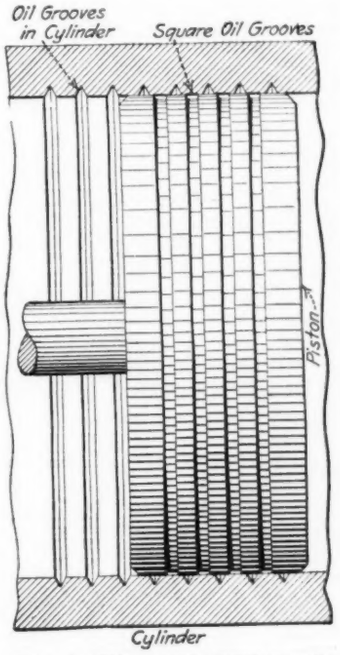


FIG. 4. EXAGGERATED VIEW OF OIL GROOVES. THE PISTON RING GROOVES ARE NOT SHOWN

We found that with old oil we got practically no lubrication on the upper 30 or 40 deg. of the cylinder. We found packing rings worn at the top and not at the bottom. By

bringing the lubrication down in from the top of the cylinder we got a drop of oil as cold as we can get it at this point. As the piston comes along it "butters" the inner circumference of the cylinder with oil.

DAILY REPORT ON VISCOSITY AND GRAVITY

We found that on a hot Summer night, the oil would "let go" and the bearings heat. After considerable experimenting we learned that this was due to two things: First, the increase in air temperature had lowered the viscosity of the oil. So we put in an oil cooler and a thermometer on each oil line. Mildly warm water is sprayed through the oil to wash it before it goes to the settling tank; it goes through a linen mesh, is filtered through flannel bags, and the engine oil is kept as near 75 deg. F. as possible.

We found that the viscosity changed practically 10 points for every 10 deg., as I remember. You put an oil into service and run it up 30 to 40 deg.—110 to 115 deg. is not an uncommon temperature on a Summer night—and you have an oil much lower in viscosity than you will buy of an oil dealer.

The oil systems are logged every hour, and once in 24 hours the viscosity and the gravity are taken. We have found this information valuable. We discovered as soon as we began recording these properties that the oil was going down every time it passed through the filter—we were taking out the heavier, or fatty, parts of the oil. Now we put the filter bags into hot water, press the heavy oil out and wash the bags; once a week we add a certain amount of stock oil, a very heavy engine oil and a plain cylinder oil. By adding a certain number of gallons every week we keep the viscosity practically constant. The oil systems are watched continually; every peculiarity is recorded. The reports come to the office every morning and are filed, and instructions are to call my attention to a drop in gravity of 1 deg. and a drop in viscosity of 10 points. To show how valuable this system is: An oiler had been using kerosene for cleaning; instead of carrying it back to the stock room he opened one of the crankcase doors of the large machine and threw in what he had left—between a pint and a quart. This became known next morning on the viscosity test. We located the man that had done it, and the viscosity was made up by the stock oil.

The viscosities of oils are measured at 100 deg. which is taken as a standard temperature. We have found that of two oils, the analyses of which were the same, one would cause the valves to drag and the other gave no such trouble. By testing the viscosity of each oil at temperatures near and up to the working temperature, we found that up to 350 deg. F. the viscosity of both oils was the same, but at a slightly higher temperature the viscosity of the troublesome oil dropped 40 points behind the other. The viscosity at working temperatures should therefore be known.

Gas-Producer Power Plants Around New York

SYNOPSIS—Operating data and the opinions of the plant owners indicated that in the majority of cases these plants are giving satisfaction. This was particularly true of the smaller plants.

"Producer Gas-Power Plants in New York City and the Metropolitan District" was the subject of a paper by C. M. Ripley before the American Society of Mechanical Engineers in New York on Tuesday evening, Nov. 9. By correspondence and personal visits the author collected data relative to the performance of most of the producer-gas plants in this section and compiled the information in the form of two tables, the first representing the opinions of the owners and the second containing operating figures.

Referring to Table 1, it will be seen that out of 33 plants approached the owners of 26 responded freely with the information sought. Whether significant or not, it appears that the responses in the case of plants under 250 hp. were more generally favorable than those from the larger plants. Sixty-five per cent. stated that repairs were not in excess of expectations, 76 per cent. had not found depreciation excessive, 73 per cent. did not consider the labor excessive, and 68 per cent. would again install producer plants if called upon to add to their present plant capacity. A singular case was that of plant No. 30, Table 2, which was bought secondhand and is giving excellent service, but which in the hands of the former owners was a complete failure.

Only two of the plants listed burn bituminous coal, four burn No. 1 buckwheat, and the majority use pea coal. In the column headed "Attendance" the plus sign after the number indicates that the services of another man are required for a few hours each day; a minus sign indicates that the attendant's whole time is not taken up with the operation of the power plant.

In the discussion following the presentation of the paper the opinion seemed to prevail that in sizes over 500-hp. the producer plant, owing to the high first cost, could no longer compete with the steam turbine with its relatively low first cost and high economy, which, in the larger units, is rapidly approaching that of the producer plant and gas engine. In the smaller sizes, however, where the load is steady and there is no demand for exhaust steam, the producer plant appears to be holding its own, even in the face of keen competition from the oil engine along the Atlantic Seaboard. In the South and Southwest, of course, the oil engine has a decided advantage, owing to the lower cost of oil and the higher cost of coal suitable for the producer.

TABLE 1. CENSUS OF OPINIONS

Question 5: Are repairs in excess of your expectations?
Question 6: Is depreciation in excess of your expectations?
Question 7: Is labor charge in excess of your expectations ?
Question 9: If you were to construct another building for similar uses, would you again install a gas-producer power plant?
Opinions vary with size.

No. of Plant	Hp. Not Answered	Hp. Answered	Ques. 5		Ques. 6		Ques. 7		Ques. 9		
			Yes	No	Yes	No	Yes	No	Yes	No	?
1		1,000	×		×		×				×
2	1,000	New plant just started									
3		700		×		×		×	×		
4		600	×		×		×			×	
5		600	×		×			×	×		
6		600		×		×		×	×		
7		400	×			×		×	×		
8	400										
9		300	×		×		×			×	
10		300	×					×	×		
11		275	×			×	×				×
12		250	×		×		×			×	
13		200		×		×		×	×		
14		200	×		×		×			×	
15		200		×		×		×	×		
16		200		×		×		×	×		
17	200										
18		180		×		×		×	×		
19		150		×		×		×	×		
20		150		×		×		×		×	
21	125										
22		150		×		×		×			
23		100		×		×		×	×		
24		100		×		×		×	×		
25	100										
26		50		×		×		×	×		
27		50		×		×		×	×		
28		50		×		×		×	×		
29	56										
30		50		×		×		×	×		
31		50		×		×	×			×	
32	40										
33		30		×		×		×	×		
Total	1,921	6,935	9	17	6	19	7	19	17	6	2
Per cent. of number	22	78	35	65	24	76	27	73	68	24	8
Per cent. 250 hp. or larger	14	86	80	20	56	44	50	50	50	30	20
Per cent. 200 hp. or smaller	26	74	6	94	6	94	12	88	86	14	0

TABLE 2. DATA ON PRODUCER POWER-PLANT PERFORMANCE

No. of Plant	Name of Co.	Producers	Total Hp. (Producers)	Engines	Total Hp. (Engines)	Dynamos	Total Kw.	Voltage	Total Cost, Dollars	Ht. Overall	Space	No. Charges Daily	Max. Load	Ave. Load	Hours Run Daily	Fuel Daily, Tons	Fuel per Hp.-Hr.	Lubricants	Attendance	Av. Repairs Yearly	Age (Years)	Kind of Coal
1	Atha Tool Co.	2	1,000	4	640	4	325	250	75,000	40 ft.	30x30*	250	†	†	12	8					5	Bit.
2	L. O. Koven Bros.	2	1,000	1	150	1	100	250		50	70x75		†	†								Bit.
3	Horton Ice Cream Co.	2	700	6	675	3	450	250 125	55,000	25	24x30*		‡	‡	24	4	1.35				5	Buck.
4	Amer. Cotton Oil Co.	3	600	2	600	2	400	125		40	22x45*	3		6,000A	24	9½					2	Pea
5	Swift & Co.	3	600	4	700	4	300	250		30	20x20*	4	‡	450A‡	24	3½			6		8	Pea
6	National Meter Co.	2	600	3	550	3	350	125	30,000	30	40x16*	1		225-kw.	11	3	1.27‖				9&3	Buck.
7	Phoenix Tube Wks.	2	400	4	535	3	300	115		20	20x20*	1			10	1.1			1		13	Pea
8	Erie R.R.	2	400	7	552	3	230	110 2,400		35	20x40*	8	§	§	24	2			3+		5&4 16	Pea
9	Castle Ice Cream Co.	1	300	1	300	1	200	125 250	18,000	20	20x30*	4	500A	750A	24	3½		90 gal. per mo.	4	$1,500	1	Pea
10	Mergenthaler Linotype Co.	1	300	1	300	1	200	125 250	33,000	Out-doors	15x15*	2			10½	2½	1.5				2	Pea
11	Max Ams Mach. Co.	2	275	2	175				20,200					142 hp.	11	¾	1.12				8	Pea
12	De La Vergne Mach. Co.	1	250	1	75	1	25	115		30	15x25*	1	§	§	11	¼					7	Buck.
13	Keystone Watch Case Co.	1	200	2	170	2	125	220	15,000	20	12x18*	21	400A	500A*	11	1½			2	$300	7	Pea
14	Auto Press Co.	1	200	1	135	1	100	240		20	14x14*	1	1,888 kw.-hr. per week	80-kw.	9	1¼			$25 per week		5	Pea
15	John Thompson Press	1	200	2	185	3	175	250		20	12x20*	8	165 hp.		11	½	1.29		$43 per week		9	Pea
16	Acme Elec. Garage	1	200	3	300	3	200	125 250	30,000	20	40x40	2	2,000 kw.-hr. per day	1,630 kw.-hr. per day	24	2			2		1&4	Buck.
17	T. Schriver & Co.	2	200	3	150	6	69	250		15	30x30				9½						8	Pea
18	Chas. Mundt	1	180	2	175	1	5	230	5,500			2	§	§	14	½	1.12		1—		8	Pea
19	Geo. F. Heinrichs	1	150	2	270	2	174	125 250		20	12x12*	8		110 kw.	16	1.1			1+		5	Pea
20	Ludwig Piano Co.	1	150	1	100	1	75	230		15	10x25*	6			11	1.1			2		7	Pea
21	Strobell & Crane	1	125	1	80	1	50	125		14	11x25	6	250A§	300A§	10	¾			1+		9	Pea
22	Hooten Cocoa Co.	1	150	1	150	1	100	220	13,000	25	25x25*	6	400A	500A	11	½	0.94	2 gal. daily	1+	$33	5	Pea
23	Hawley & Hoops	1	100	1	80	1	75	230		14	10x20*	5			10½	½	1.12		1		9	Pea
24	Winelander & Jackson	1	100	1	85	1	15	110		18	20x20*	1	§	§	12	½		2 qts. daily	1—	$79	4	Pea
25	Universal Metal Bed Co.	1	100	1	100	1	¼	115		17	26x16*		§	§	10	½			1		8	Pea
26	Thos. Wright Wagon Shop	1	50	1	35	1	7	115	3,640	15	12x30	4	§	§	9½	½		$2 per mo.	1—	$52	9	Pea
27	McCabe Hanger Co.	1	50	1	50	1	30	250		15	25x25	4	180	120	10	⅜			$70 per mo.		8	Pea
28	Pirika Choc. Co.	1	50	1	45	1	10	115	4,200	Out-doors	15x15*	1	‡	‡	11	¼	1.12	$15 per year	1—	$10	9	Pea
29	Erie R.R.	1	56	1	56				Sec. hand	25	20x30		§	§		¼			1—		3	Pea
30	Leiman Bros.	1	50	1	40	1	5	115	Sec. hand	12	12x12*	3			11	¾			1—		8	Pea
31	N. Drake Co.	1	50	1	50					18	10x25*	1			11	¼			1—		9	Pea
32	Allsopp Bros.	1	40	1	40					12	6x15*	3	§	§	11	¼			1		7	Pea
33	Gen'l Cement Prod. Co.	1	30	1	28	1	18	115	3,300	15	10x10	1	§	50A§	24	½		$5 per mo.	1	$70	6	Pea
	Experimental																					
	Stevens Institute	1	30	3	29	1	7½	125														
	Stand. Motor Const. Co.	2	300																		7	
	Furnaces Only																					
	Cork Insulating Co.	2	400										†		24	12			2		2	½ pea ½ buck.
	Hyatt Roller Bearing Co.	3	1,500										†								1	Bit.
	Crucible Steel Co.	2											†									
	Producers Discontinued																					
	N. J. Adamant Mfg. Co.	1	50	1	50				Sold												3	Pea
	Harper Brick Co.	1	27	1	27				Sold												2	Pea
	Newark Spring Mattress Co.	1	50	1	38				Sold to Leiman Bros.												4	Pea
	C. E. Hertlein	2	285	3	300				Sold 2 plants, both failed. Now use steam elect.												1	Pea
	Cameron Mach. Co.	1	50	1	50				Sold						9	½			1		3	Pea
	Manhattan Screw & Stamp Works						1	75	For sale. Firm retired from business.													
	Adriance Machine Co.								Second hand. Never unboxed. For sale.												2	

* Space for producers only. † Furnace gas load. ‡ Belted refrigeration load. § Belted power, pumps, compressors, etc. ‖ At full load.

Boiler-Furnace Dimensions

At the thirty-second meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers, held at Zanesville, Nov. 18 and 19, Osborn Monnett, ex-chief of the Chicago Smoke Inspection Department, read a paper dealing with ways to improve conventional boiler settings to reduce smoke where high-volatile long-flaming coal is used.

Instead of the 6-ft. setting with chain-grate stokers and vertically baffled boilers, Mr. Monnett finds that a 10-ft. setting with long combustion arch, full extension and leaving a throat of 30 in. between the end of the arch and the bridge-wall, is better. The dimensions of the throat may, however, be varied from 18 to 36 in., depending upon the local conditions. With chain-grate settings, the height of the arch above the grate formerly was 11 in. but is now 15 in.; the slope of the arch has been increased to 2 or 3 in. per ft. Where the arch is sprung across the furnace it is set low, 9 in. above the grate at the skewback and with a 9-in. spring, giving a height at the center of 18-in. from the grate. The horizontal baffles are always good provided 6 ft. 6 in. is the minimum headroom allowed; that is with a boiler like the Babcock & Wilcox, for example, 6 ft. 6 in. between the bottom of the front headers and the floor line.

With double inclined stokers and boilers like the foregoing, Mr. Monnett recommends the 10-ft. headroom. The furnace itself is of course extended. With front-feed stokers a clearance of 7 ft. is not sufficient to give good results, but 10 ft. will be found satisfactory. It may be necessary to excavate to get this headroom, but this usually is easily done. With underfeed stokers of the Jones and American types, 8 ft. 6 in. headroom for vertically baffled boilers gives good results, and for horizontally baffled boilers 7 ft. will be sufficient. With tubular boilers the stokers may be installed with 42 in. from the bedplate to the shell. With vertically baffled boilers having stokers of the Taylor type, 10-ft. headroom gives excellent results.

With hand-fired furnaces the common flush-front 24-in. setting gives, as is well known, very smoky results. From a smoke standpoint, the old design of Dutch oven is poor. Contrary to stoker practice where the fuel is introduced slowly and in small quantities, there is considerable coal thrown on the fire at once. The strong radiation from the brickwork above the fire has the effect of distilling the gases so rapidly that puffs of dense smoke will be made after firing in spite of every effort to prevent them. By putting in an arch immediately over the bridge-wall and close to the boiler shell, extending it back some distance and putting immediately back of it a single-span deflection arch, a hand-fired tubular boiler with 54 in. from the bottom of the boiler to the floor will give good results from the smoke standpoint.

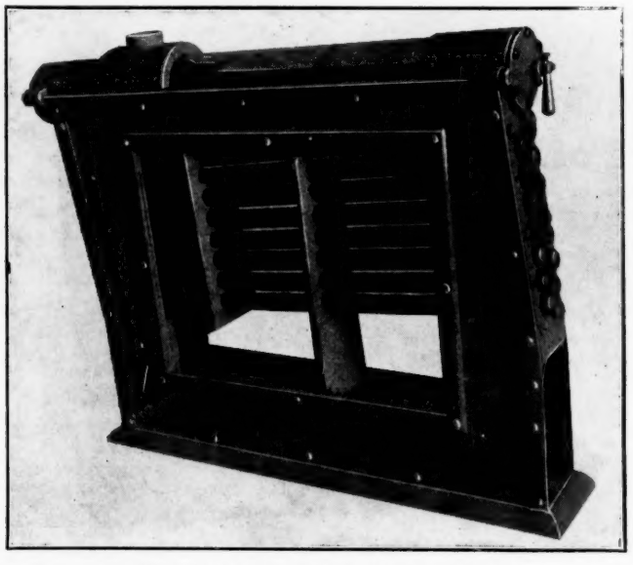
It is not practical to combine a hand-fired coal-burning furnace with a vertically baffled water-tube boiler, but it is a simple matter to arrange such a furnace with a horizontal baffle, carrying out the same idea as for the hand-fired setting just described. The ordinary hand-fired horizontally baffled water-tube boiler furnace is covered with box tile and has all the defects of the now obsolete type of Dutch oven, as

it is practically a firebrick inclosed furnace from which the volatile gases will be distilled at a rapid rate. To avoid this condition the first two rows of tubes should be bared to the fire by putting T-tile on the second row and by using a two-span deflection arch to break up the current of the gases. There should be from 20 to 25 per cent. of the grate surface in free opening above the bridge-wall. The free opening from the back of the bridge-wall to the deflection arch should not be less than 40 per cent. of the grate surface, while the free opening under the deflection arch should be 50 per cent. of the grate surface. Hand-fired furnaces for high-pressure work should be fitted with four air-siphon steam jets placed across the furnace above the fire-doors and used when necessary.

Mr. Monnett here went into a discussion of furnaces for low-pressure heating plants.

ENGINEERING AFFAIRS

American Institute of Steam Boiler Inspectors—At a recent meeting of this society in New York City, A. C. Lippincott, of the E. Keeler Co., makers of boilers, demonstrated



GLASS MODEL OF KEELER BOILER

the circulation in the Keeler boiler by using a glass model. Discussion and comment followed the demonstration. Mr. Lippincott will give other demonstrations before organizations interested.

The Combined Engineers of Brooklyn, N. Y., held their eighth annual entertainment and reception at Kismet Temple on Saturday, Nov. 20. The attendance, as usual, was large, and an enjoyable vaudeville performance preceded the dancing. The officers of the combination are: S. A. Wright, chairman; Charles A. Enggren, vice-chairman; Walter F. Brundage, secretary; John P. Martin, financial secretary; George O. Kaley, treasurer; Charles Baxter, sergeant-at-arms. The associations comprising the combined organization include Brooklyn Associations No. 8, No. 27, No. 41 and No. 57 of the N. A. S. E., and the Modern Science Club.

A Heating and Ventilating Correspondence School—A correspondence course prepared especially for those who are already in the heating business or allied lines, limited to the design of direct heating systems and completed in ten lessons, has been prepared by the New York Correspondence School of Heating and Ventilation, 132 Nassau St. This course is available for those who are unable to attend day or evening classes. Upon its completion the student will be qualified to take up the further study of ventilation and power-plant design. The fee for this course will be $10, payable in advance. Entries may be made at any time, and the student may progress as rapidly as he wishes.

The First Railroad in the Mississippi Valley, called the Coal Mine Bluffs R.R., was built in 1837 for the purpose of delivering coal from the bluffs of St. Clair County, Illinois, to St. Louis. The road was six miles long and was characterized at the time as the greatest engineering enterprise ever consummated in Illinois.

NEW PUBLICATIONS

EXAMPLES IN ALTERNATING CURRENTS. Vol. I. By Prof. F. E. Austin. Published by the author at Hanover, N. H. 1915. Size, 5x7½ in.; 223 pages; illustrated. Price, $2.40.

A well-selected list of 57 problems covering electrical pressures, sine curves, inductance, impedance and capacity. Trigonometry and calculus are used freely in the solution of the problems, and for a handy reference the first 45 pages are devoted to a review of these mathematical functions. A number of useful tables and diagrams are also included. The book will find its greatest use among students and engineers who have had some training in mathematics and the fundamentals of electricity.

HOW TO MAKE A TRANSFORMER FOR LOW PRESSURES. By Prof. F. E. Austin. Second edition. Published by the author at Hanover, N. H. Size, 7x4½ in.; 17 pages; illustrated. Price, 40c.

It frequently happens that the operating man, student or amateur electrician desires to construct a small transformer to operate on a 110-volt lighting circuit and produce a low secondary voltage for the operation of bells, signals or alarm circuits. This little booklet contains full directions for making such a transformer, with suggestions for different coil connections so as to produce various secondary voltages. It is excellent in so far as telling "how to do the work," but the amateur perhaps would be better satisfied had the text also told "why" the particular dimensions and sizes were selected; that is, preliminary calculations showing the method of determining these items might have enhanced the usefulness of the book.

MODERN BOILER-ROOM PRACTICE AND SMOKE ABATEMENT. By James T. Hodgson. Published by "The Railway Engineer," London, England. Cloth; 8x5 in.; 313 pages; 209 illustrations. Price, 3s. 6d.

Being an English book, it treats chiefly of English practice and equipment. There are twenty short chapters in 10-point type on a good grade of white paper. The first chapter is a general introduction to the subject. Chapters II-VII treat of combustion practice and apparatus related to it, as draft, hand-firing methods, mechanical stokers, coal- and ash-conveying systems and flue-gas analysis.

Chapters VIII and IX take up the construction of boilers and boiler mountings and accessories. Softening and purification of feed water are the subject of Chapter X, and feed-water heating and pumping are treated of in Chapters XI and XII. The properties and calorimetry of saturated and superheated steam are the subjects of Chapters XIII and XIV. Chapter XV relates to steam pipes and their fittings. In Chapter XVI the author has many good suggestions on the general supervision of the boiler room. The operating engineer will find this the most serviceable chapter in the book. The definitions and equivalents of technical terms in Chapter XVII will help the beginner in the field. The three pages comprising Chapter XVIII contain instructions to boiler attendants. These are as complete as the small number of pages used will permit, though one cannot but wish after reading them that the author had devoted more space to the subject in just this way.

The last two chapters will be well received, as they are devoted to questions and answers on the most important of the boiler-room engineer's work.

A. S. M. E. BOILER CODE—NEW EDITION

The American Society of Mechanical Engineers has recently issued a new edition of the Boiler Code containing a comprehensive index. This is divided into two parts—one a general index to the complete rules and the other containing sectional indexes to the parts referring to New Installations of Power Boilers, New Installations of Heating Boilers, and Existing Installations. Copies of the new edition can be obtained from the society, 29 West Thirty-ninth street, New York City.

TRADE CATALOGS

Detroit Stoker Co., Detroit, Mich. Catalog. "V" type side feed stokers. Illustrated, 26 pp., 8½x11 in.

Mesta Machine Co., Pittsburgh, Penn. Bulletin R. Barometric condensers. Illustrated, 8 pp., 6x9 in.

M. Smolensky Mfg. Co., Cleveland, Ohio. Catalog. Suction and check valves. Illustrated, 16 pp., 6x9 in.

Edge Moor Iron Co., Edge Moor, Del. Catalog No. 52. Water tube boilers. Illustrated, 114 pp., 6x9 in.